

# COMPRESSED AIR SYSTEM REVIEW

Prepared for

**INTERMOUNTAIN POWER SERVICE CORP.**

**Intermountain Power Service Corp.**

**850 W. Brush Wellman Road  
Delta, UT 84624-9522**

**February 2008**

Prepared by



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**IP12\_006149**

# Intermountain Power Service Corporation

## Compressed Air System Review: Executive Summary

Intermountain Power Service Corporation currently spends \$775,071 annually on energy to operate the compressed air system at Delta, Utah. This figure will increase as electric rates are raised from their current average of 5.0 cents per kWh. The set of projects recommended below could reduce these energy costs by \$385,214 or 50%. Estimated costs for completing the projects cannot be finalized until IPS obtains project cost quotes, but it is expected that the total cost will yield a project with less than a 2-year payback or \$770K.

| PROJECT   | SAVINGS PROFILE                                  | ENERGY AND OTHER SAVINGS  |           |                      | TOTAL PROJECT COST (\$)            |
|---|--|---|-----------|----------------------|------------------------------------|
|   |  | AVG kW  | kWh       | SAVINGS (\$)         |                                    |
| AIR COMPRESSOR SUPPLY   |  |   |           |                      |                                    |
| 1. Reduce discharge pressure of compressor from 125 psig to 110 psig – increase flow 7% and increase turndown – 157 scfm each or 628 scfm, 4 compressors                            | Additional air flow using same amount of energy. |   |           |                      | Part of Projects #2 , #6,and #7    |
|   | 314 scfm (equivalent new capacity)               | 73.0  | 639,800   | \$31,990             |                                    |
| CAPACITY CONTROL  |  |   |           |                      |                                    |
| 2. Add central master control system with inlet guide vanes allow full turndown to meet conditions (15%) and lower electrical energy use 85%  | 208 kW   | 208   | 1,822,080 | \$91,104             | TBD                                |
| AIR TREATMENT   |  |   |           |                      |                                    |
| 3. Replace four current heatless dryers with new modern similar-sized blower purge dryer with auto dewpoint demand controller   | 1348 scfm  | 257.4   | 2,254,680 | \$113,946 (\$1,212)* | Est. Cost \$220,000 + installation |
| 4. Replace notched ball valve drains with level-activated drains (on 4 main water-cooled after-coolers)--2 drains per point of each compressor aftercooler separator (eight drains) | 110 cfm  | 21.2  | 185,960   | \$9,298              | \$9,350 installed                  |
| DEMAND-SIDE SYSTEM  |  |   |           |                      |                                    |
| 5. Remove orifice plates on receiver entry to the casings on Units #1 and #2 Bag House and install appropriate regulators on the discharge line of each receiver.                   | After dryer 10 – 20 psig                         | This is not an energy issue but a reliability and production issues |           |                      |                                    |

| PROJECT   | SAVINGS PROFILE  | ENERGY AND OTHER SAVINGS |               |                    | TOTAL PROJECT COST (\$) |
|---|--|--------------------------|---------------|--------------------|-------------------------|
|   |  | AVG kW                   | kWh           | SAVINGS (\$)       |                         |
| 6. Eliminate excessive pressure loss in compressor area between compressor discharge and distribution system, reconfigure piping systems as required. | 481 scfm<br>(16 psig)  | 92.8                     | 813,180       | \$40,659           | TBD                     |
|   | This is not only a direct energy issue but a project to enhance reliability and production |                          |               |                    |                         |
| 7. Implement ongoing leak management program  | 855 scfm   | 165.0                    | 1,445,460     | \$72,273           | \$20,600                |
| <b>Special Project</b><br>8. Replace air operated air horns used in warm weather with electric-operated units with same flow;                         | 124 scfm per horn<br>Equivalent of 10 horns<br>2190 hrs (summer season)                    | 248.0<br>(2190 hrs)      | 543,120       | \$27,156 / yr      | \$20,000                |
| <b>TOTAL</b>  |  | 1,065.4 kW               | 7,704,280 kWh | \$385,214 per year | <\$770K (<2-yr payback) |

\* Added electric cost from new dryer use of electricity directly.

Savings estimates depend, in part, on the capacity control system effectively translating lower air use into reduced electric cost. The current system does not have this type of unloading controls. With today's piping system, the controls will not accomplish this goal.

It also is important to note that other recoverable compressed air costs can also be considered, e.g., air system maintenance, water costs, and equipment life. Usually, the electric cost is between 50% and 75% of the total "variable compressed air costs." Associated maintenance and other costs are often more than 30% of the identified electric cost.

**THESE PROJECTS AND THEIR PERFORMANCE ARE INTERCONNECTED. IF ALL THE PROJECTS ARE NOT IMPLEMENTED, THEN VERY LITTLE OF THE SPECIFIC PROJECTED SAVINGS WILL MATERIALIZE.**

#### PROPOSED ACTION PLAN

- Reconfigure piping from dryers to main header eliminating 16 psig of pressure loss.
- Remove six orifice plates at feeds to Unit #1 and Unit #2 Bag House air receivers; replace with appropriate regulators on discharge of receivers.
- Replace heatless dryers with new similarly sized blower purge dryers.
- Install inlet guide vanes on all four compressors with compressed air central air management system.
- Replace notched ball valve drains on the aftercoolers separator drains with appropriate level activated electric or pneumatic actuated drains.
- Repair tagged leaks, implement a continuing leak repair program.
- Special: replace air operated cooling air horns use in the summer with electric operated, avoid 1240 scfm use / 5000 hours per year.

- After the system is reconfigured and stabilized, review the benefits in moving the air compressor inlet to receive cleaner dryer and cooler outside air.
  - This will have a positive impact on maintenance costs.
  - During the colder months up to another 150 scfm will be available assuming the motor can handle the 6% increase in power. This extra air may allow you to keep a third unit off line.
  - We do not recommend doing this unless the plant installs the proposed CEC control system with inlet guide vanes or something else equal.

## PHASE 2 ACTIVITIES

After the basic system reconfiguration is implemented, a continuing review of the plant is recommended:

- Review regulator operations to ensure they are working and at lowest effective pressure.
- Review air-operated diaphragm pump with possible switch to electric-operated units if larger units or high cycles become the practice.
- Review the opportunity and payback of replacing older motors with high-efficiency units, whenever major electric motor repair is anticipated.
- Continue an aggressive leak tagging and repair program; quantify and value the leaks and report to management on a predetermined regular basis.
- After the system is reconfigured and stabilized, continue to review all dust collector installations.
- Continue to look for air-operated vibrator applications; monitor incoming equipment.

# COMPRESSED AIR SYSTEM REVIEW

Prepared for

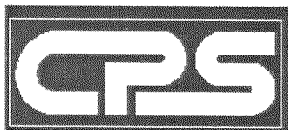


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\*Disclaimer: This report provides a general overview of the facility's compressed air system. As such, all data and analysis presented are estimates and should be only considered as guidelines. Final project specification and enumeration of potential savings and costs should be developed using appropriate compressed air system professionals. Cost and savings estimates and "totals" included in tables may reflect rounding.

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## CHAPTER 1. COMPRESSED AIR SYSTEM REVIEW – OBJECTIVES

The REPORT SECTION of the Compressed Air System Review identifies specific projects to reduce air usage. These reductions usually translate into lower electric costs, improved system operation, and enhanced air quality and productivity. For a summary of results for this section, refer to the EXECUTIVE OVERVIEW at the front of this notebook.

For details of data gathered and work sheets completed, refer to the PLANT SURVEY SECTION of the notebook. For equipment performance and details, see the EQUIPMENT SECTION. For project cost estimates, refer to the PROJECT COST SECTION. For additional information and articles, see the MISCELLANEOUS SECTION.

The primary objective of the review is to provide a comprehensive list of specific measures needed to improve compressed air system operation and cost-effectiveness in the short- and long-term. The review addresses these topics:

- Review appropriateness of major equipment pieces in the compressed air system to produce the right quality and quantity of usable compressed air at an acceptable efficiency
- Develop a load profile of compressed air production
- Identify current electric power cost per cfm in order to establish a baseline for evaluating potential projects
- Evaluate characteristics and appropriateness of the use of a central compressed air capacity control system
- Outline plans for an ongoing leak management program
- Identify savings potential in use of air saving devices such as nozzles and auto drains
- Identify savings potential in replacement or re-evaluation of “potentially misapplied air” such as cabinet coolers, vacuum pumps, and bearing cooling
- Identify critical areas, if any, to utilize planned storage in the system:
  - \* Create effective storage, if required, for capacity controls
  - \* Establish stored volume to offset identified peak demand — local in system or off system
  - \* Establish stored volume to help set up proper use of pressure/flow controller
  - \* Create effective demand-side storage, as required, at critical points
- Estimate benefits of recommended savings measures, including reduced electric consumption and maintenance costs and improved productivity and system operation.

## CHAPTER 2. CURRENT AND PROPOSED SYSTEM REVIEW

### 2.1 CURRENT SYSTEM BACKGROUND

The power plant at Delta, UT, owned and operated by Intermountain Power Service Corporation, is a coal-fired plant with two basic power generating units. The compressed air supply is anchored by four 700-hp (660-hp) class, 3-stage centrifugal compressors with a rated flow of 2,242 scfm each at 100 psig. The compressed air then goes to four Pall Trinity (Pneumatic Products) heatless-type, twin tower desiccant dryers rated for a similar flow at 100 psig, 100°F inlet air.

The compressed air is used for instrument air throughout the complex and service air. The areas to be investigated are:

- The baghouse water and air flow – calculated cylinder air – leaks – air horn – evaluate piping – orifice restriction plates, etc.
- The appropriate use of compressed air dryer with regard to operating cost and required pressure dewpoint.
- Lime preparation compressed air facility.
- Water treatment compressed air supply.
- Boiler control air compressed air supply.
- Stack compressed air supply.
- Coal car unloading.
- Sludge transfer.
- Scrubber compressed air supply.
- Central air manifold.
- Crusher building.
- Coal yard transfer buildings.
- Limestone unloading.

Overall, the basic air system review is to:

- Document where reasonably feasible where the compressed air is being used and specific air where air conservation actions would be meaningful.
- Identify and tag compressed air leaks throughout the system.
- Analyze the dryer for the compressed air with regard to performance capability of supply reliable production and recommend any replacement or upgrades that may be in order.
- Evaluate the current FS/Elliott 3-stage centrifugals in a similar manner.
- Review the impact of compressed air inlet location – second floor from the area where the dryers are located.
- Review opportunities that may exist from unit performance upgrades and/or modern microprocessor control systems, etc.

## Setting the Baseline

The following actions were taken to establish the baseline for flow and pressure.

1. Temperature readings were taken on all units with an infrared surface pyrometer. These were observed and recorded to relate to the unit's performance, load conditions and integrity. The findings were recorded on the table of compressor supply operating data that follows.
2. Critical pressures including inlet and discharge were measured with Ashcroft digital calibrated vacuum and pressure test gauges with an extremely high degree of repeatability. Findings were also recorded in the table of appropriate compressor supply operating data specific pressures were taken and logged at points (see drawing). Plant personnel measured and logged unit amperage, voltage, and power factor simultaneously and operating kW was calculated from this data and compared to OEM data at the same discharge pressure.
3. Flows were measured after the compressed air dryers with heated wire-type, thermal mass flow meters and logged with MDL multi-line loggers. Results were compared to the OEM-supplied original test data and corrected for location conditions at the measured pressure operating times. Like most electric power plants, the Intermountain Delta plant runs 24 hours a day, seven days a week, or 365 days per year except for planned outages.
4. The normal full production includes running three to four compressors and four compressed air dryers at full load for 8,760 hours per year. The cost of compressed air to calculate and reduction paybacks is \$50 per megawatt or \$0.05 kWh.



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Ambient Pressure: 12.4 psia  
 Ambient Temp: 35°F  
 Ambient RH: 30%  
 Room Temp: 75°F  
 Date: 5 February 2008  
 Shift: First/Second Shifts

### CENTRIFUGAL COMPRESSOR MEASURED OPERATING DATA

| Compressor Unit   | 1D             |       | 1C             |      | 1B     | 1A             |      |
|---|----------------|-------|----------------|------|--------|----------------|------|
| Model   | 310DA3         |       | 310DA3         |      | 310DA3 | 310DA3         |      |
| Inlet Air Temp °F   | 95°            |       | 90°            |      | --     | 87°            |      |
| Inlet psia (estimated)  | 12             |       | 12             |      |        | 12             |      |
| FL Flow (scfm)  | 2,242          |       | 2,242          |      |        | 2,242          |      |
| Capacity Control Type   | IBV/BOV        |       | IBV/BOV        |      |        | IBV/BOV        |      |
| Discharge Pressure (PG) (psig)  | 119**          |       | 119**          |      |        | 119**          |      |
| Discharge Pressure (TG) (psig)  | 119            |       | 119            |      |        | 119            |      |
| Nominal Set Point (psig)  | 120            |       | 120            |      |        | 125            |      |
| Amp Limiters Set Points (min ____ / max. ____ )   | --             | --    | --             | --   |        | --             | --   |
| H <sub>2</sub> O In °F / H <sub>2</sub> O Out °F Unit   | 76.8           | 99.6  | 76.5           | 99.4 |        | 73             | 98.3 |
| Air Temp 2 <sup>nd</sup> Stage In/Out °F  | 121            | 295   | 136            | 255  |        | 130            | 280  |
| Air Temp 3 <sup>rd</sup> Stage In/Out °F  | 121            | 250   | 120            | 228  |        | 118            | 265  |
| Calculated Full Load kW/amps  | 522.12 / 50.25 |       | 522.12 / 50.25 |      |        | 522.12 / 50.75 |      |
| Measured Full Load kW   | --             |       | --             |      |        | --             |      |
| Percent Open IBVposition/BOV*/temp  | Open           | 153°F | Closed         | 90°F |        | Closed         | 81°F |
| Percent Open IGV  | --             | --    | --             | --   |        | --             | --   |
| Estimated Average Flow inc blowoff  | 2,242          |       | 2,242          |      | 2,242  |                |      |
| PKG Disch Air Temp °F   | 221°           |       | 258°           |      | 248°   |                |      |
| General Comments:<br>Two heatless dryers were off line – 674 scfm (purge not used) and dryers were in bypass (less pressure loss).<br>IBVs are all full open – IC is 10-15° off wide open but not moving.<br>*Amperage calculated at 6,600 volts/90 PF. |                |       |                |      |        |                |      |
| After-cooler  |                |       |                |      |        |                |      |
| H <sub>2</sub> O In °F / H <sub>2</sub> O Out °F  | 73             | 82    | 73             | 86   | OFF    | 248            | 83   |
| Air In °F / Air Out °F  | 221            | 84.2  | 258            | 83   |        | 77.4           | 81   |

Figure 1. IPSC Control Air / Non-Essential

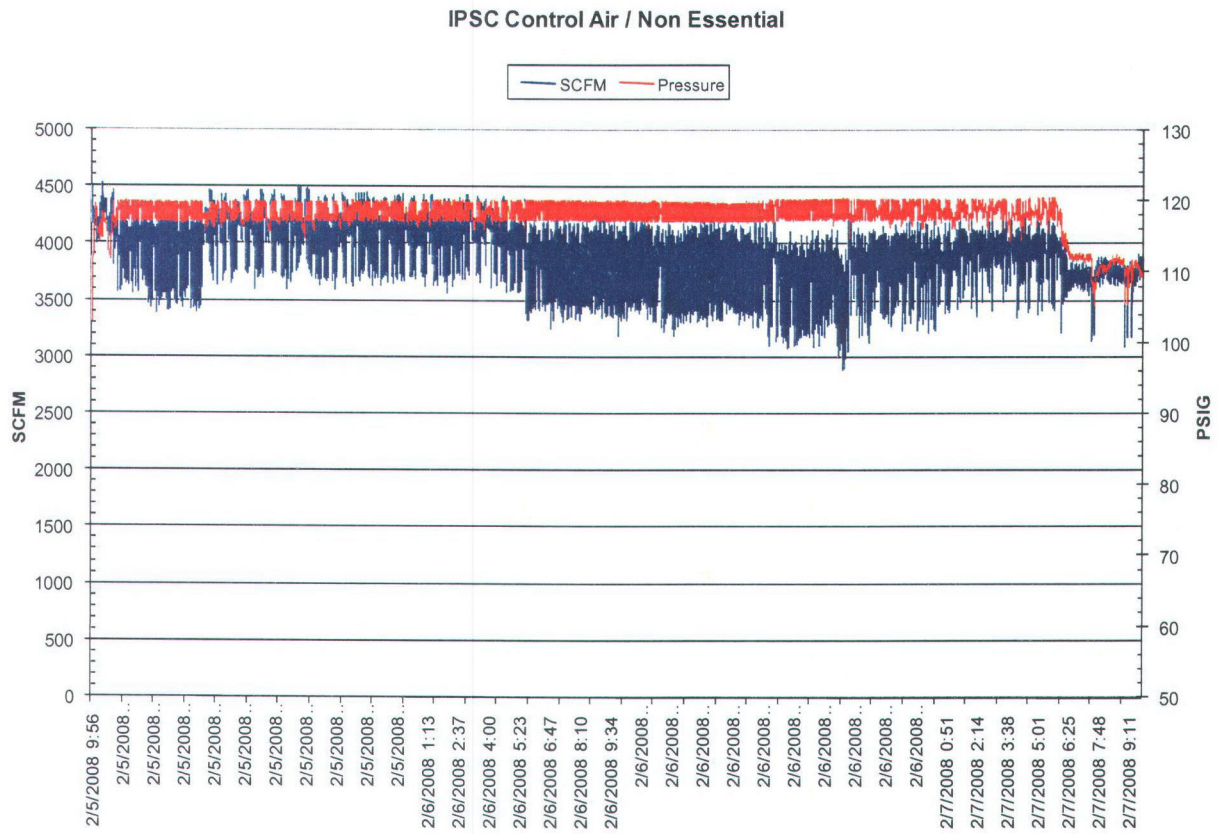
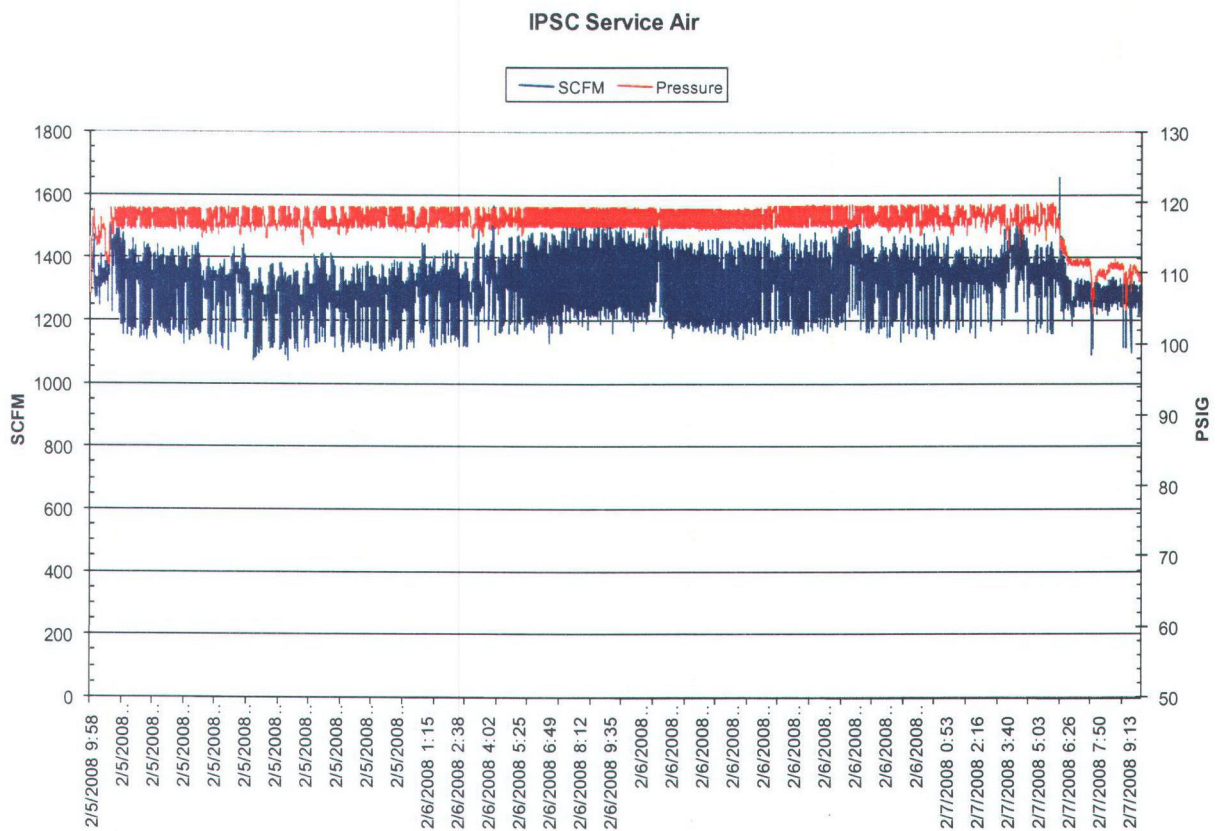
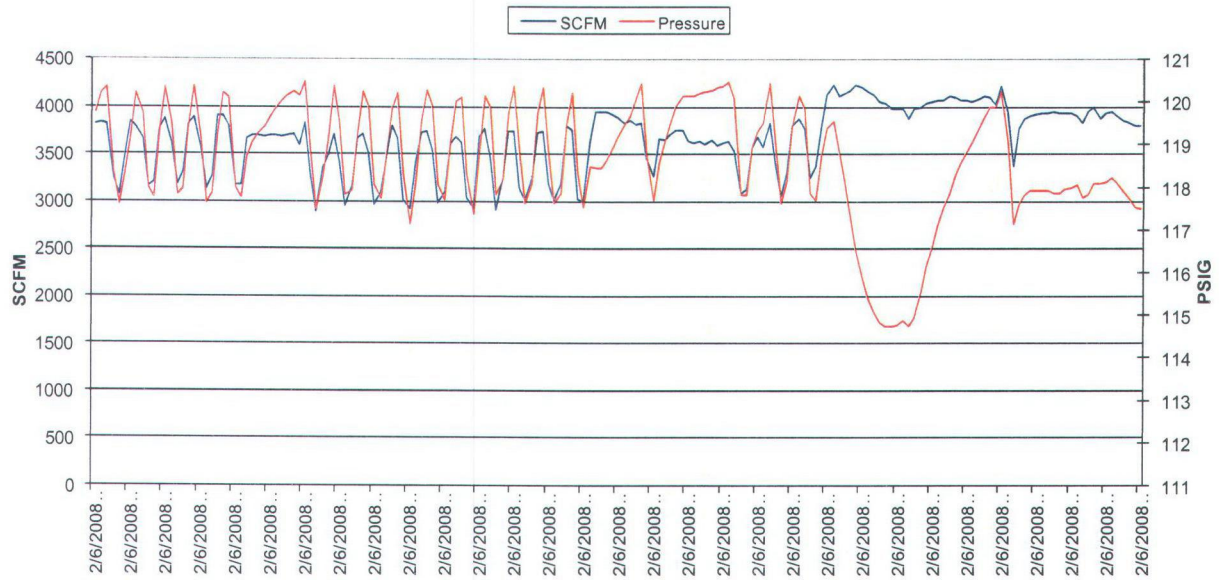


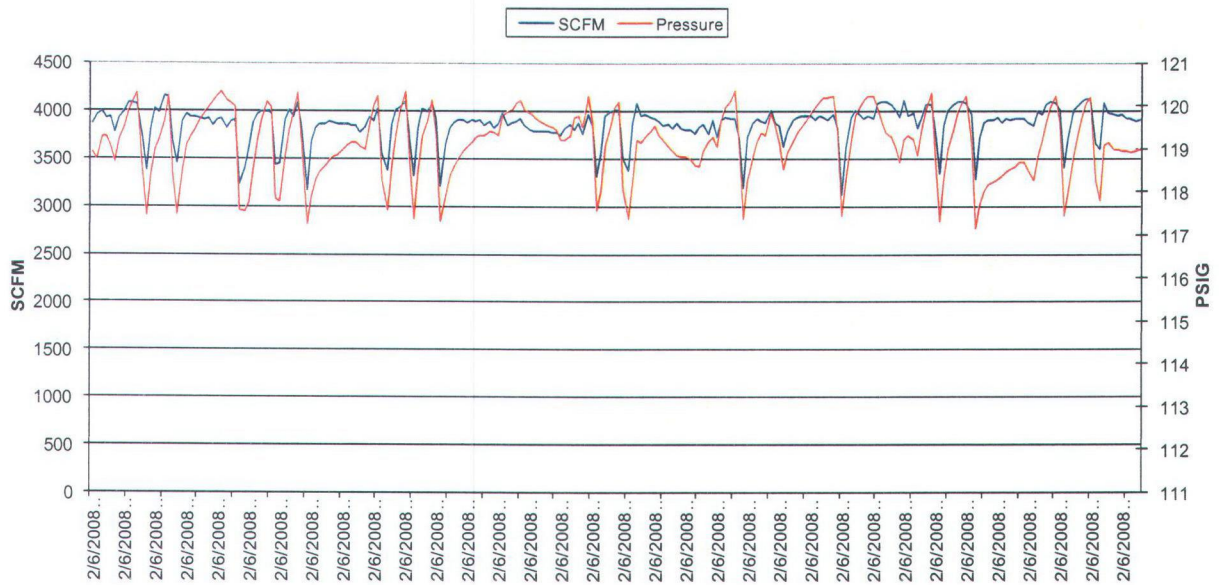
Figure 2. IPSC Service Air



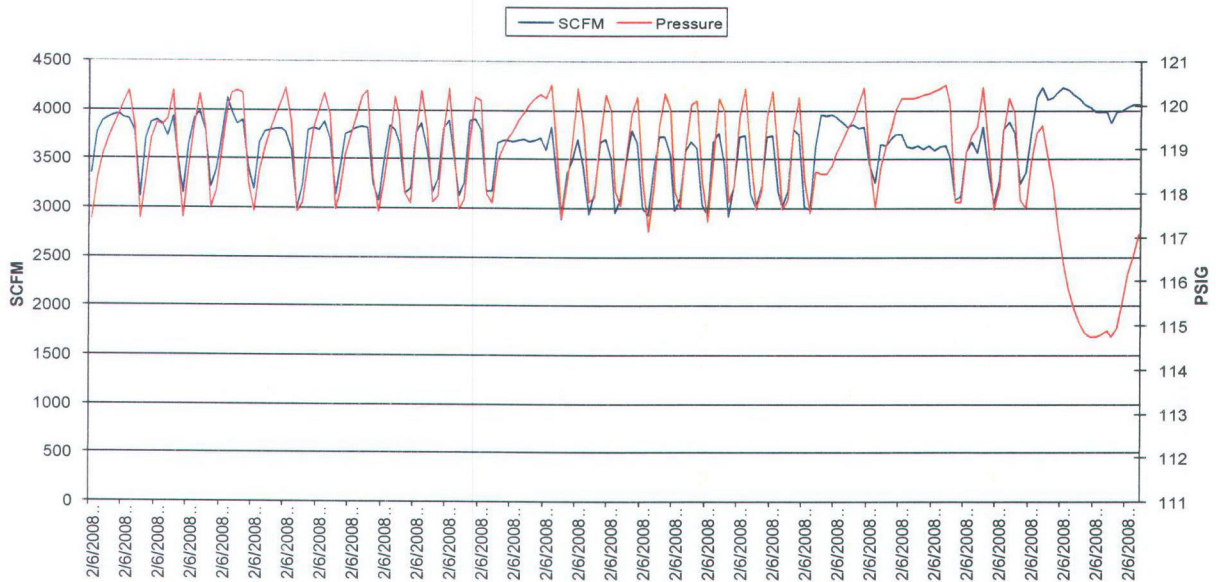
**Figure 3. Coal Yard Flow  
Isolated from 2008 to 2025**



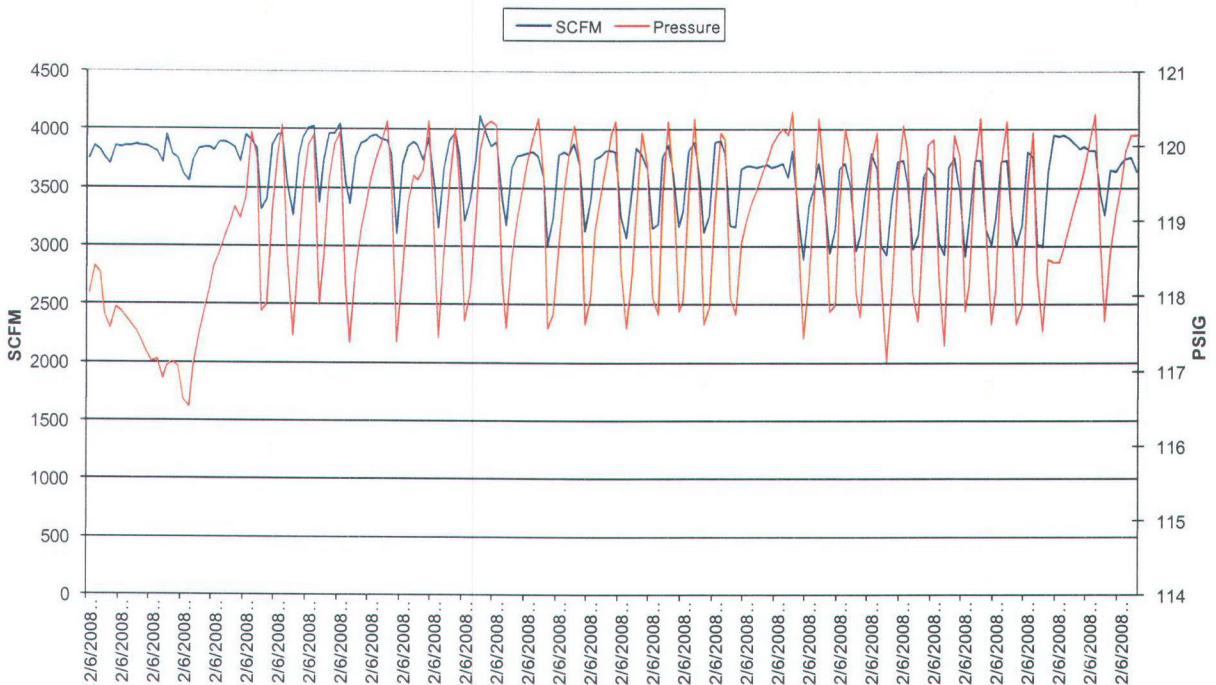
**Figure 4. Water House Flow 2  
Isolated Slaker Air Ejectors  
Isolated from 2055 to 2122**



**Figure 5. Water House Flow 1  
Isolated from 2000 to 2018**



**Figure 6. Sludge Flow  
Isolated from 1955 to 2012**



The preceding chart shows the operational data of the three 700-hp class Elliott centrifugal units had while running from 9:00 a.m. to 10:00 a.m. This data reflect the operation with two dryers off, and therefore, 674 scfm of purge air demand, which is normally on the system – was not!

There are no observable indicators for the BOV action, but measuring temperature of the blow off manifold is an excellent indicator. As the high temperature discharge air enters the blow off, it raises the temperature of the line and manifold.

Reviewing the highlighted table, you will see that Units #1A and #1C have manifold temperatures consistent (82°F / 90°F) with the pipe temperature and the ambient indicating the presence of little or no hot bypass air. On the other hand, Compressor #1D has a significantly higher manifold temperature (153°F), indicating a continuing flow of hot bypass air.

Running the Bag House flow test to determine the affect of the air horns:

- Compressor 1D    IBV    Full open    Blow off manifold temp = 153°F  
                      BOV    Open            1,342 scfm
- Compressor 1C    IBV    Full open    Blow off manifold temp = 90°F  
                      BOV    Closed
- Compressor 1B    OFF
- Compressor 1A    IBV    Full open    Blow off manifold temp = 82°F  
                      BOV    Closed

Referring to the following drawing of the basic piping system serving the Bag Houses:

- The air supply line to the non-essential air was shut off. All metered flow air was going to the bag house. Average non-essential air flow is 1,165 scfm.
- The bag houses were run with the air horns on. Each horn runs 10 seconds approximately every 4 ½ minutes. The air horns are computer controlled. There were three sets of air horns operating in the two units. Average air demand is 2,863 scfm with air horns on.
- The bag houses were then run without the air horns operating. The flow volume fell to an insignificant level.
- The service air remained a very steady 1,365 scfm.

During the Bag House test (Units #1 and #2), the compressors operation was observed:

- Total air flow on line was 6,735 scfm.
- 9:20 a.m. to 9:30 a.m. – system ran supplying all air to the Bag House, non-essential, and service air:
- Total air flow: FM1 (4,028 scfm) and FM2 (1,364 scfm) = 5,393 scfm.
- 9:31 a.m., we shut off the valve feeding the non-essential demand (see line drawing). FM1 fell to 2,863 while FM2 remained the same.

## Conclusions (see charts)

- Non-essential air at this time had a value of 1,165 scfm (4,028 – 2,863).
- The 1,165 scfm reduction did not unload compressors 1A or 1C. It did increase the blow off in Compressor 1D to almost 100% flow.
- Blow off manifold temperature was 240°F.

## Other Conditions

Two of the four desiccant dryers were off and bypassed. This had the following effect:

- With four dryers running, another 674 scfm will be required from the compressor to supply the purge air for the two dryers.
- The pressure loss, which with the bypass is negligible, will become more significant.
- Utilized properly, the dryers can begin to deliver dry air to the system to evaporate the condensate loaded in while they were out of service.

**Figure 7. IPSC Delta, UT**

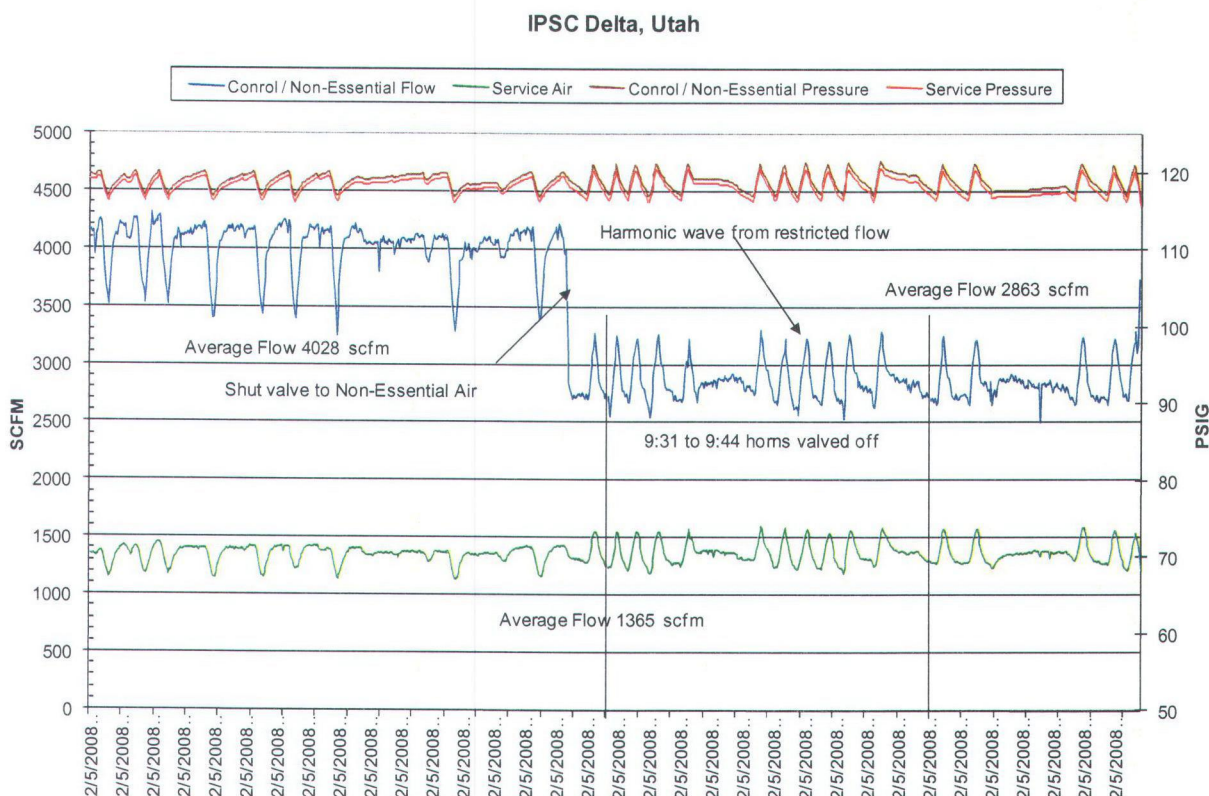


Figure 8. Bag House Test: Metering 9:10 a.m. to 9:43 a.m. 02/05/2008

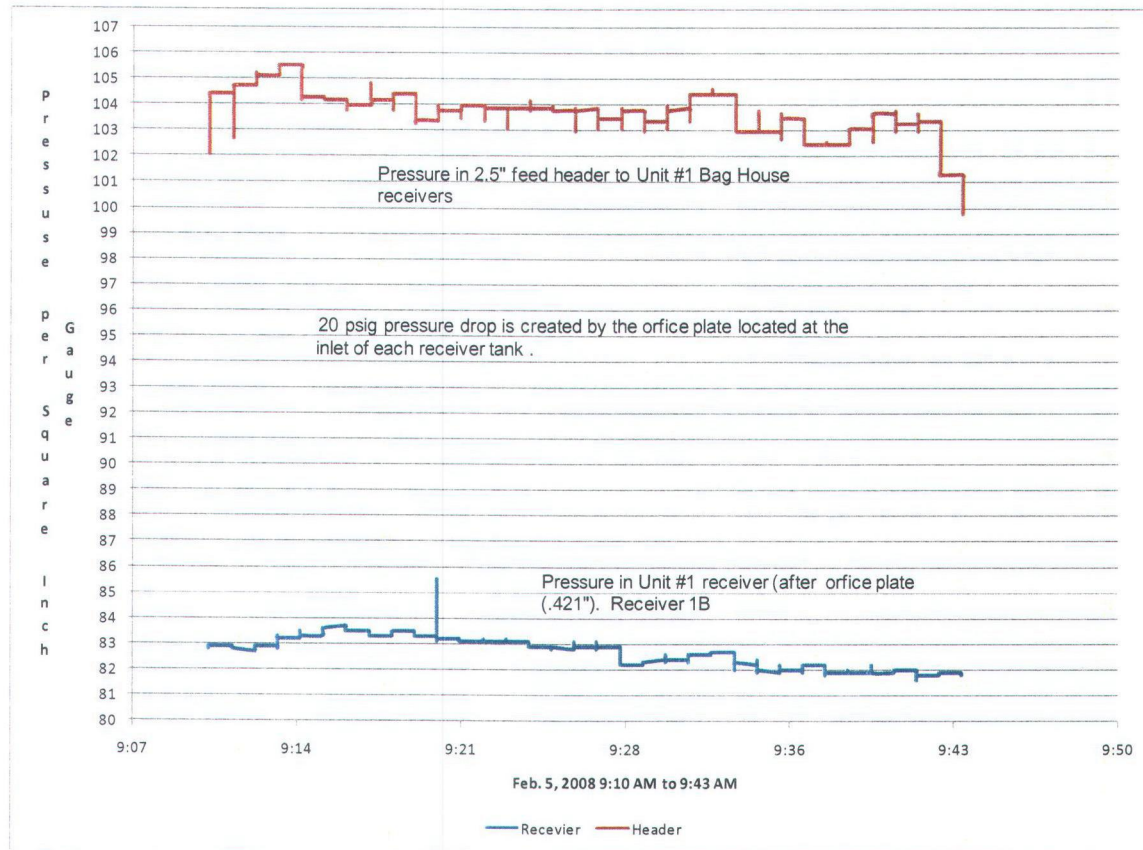


Figure 9. Unit #1 Receiver Pressure

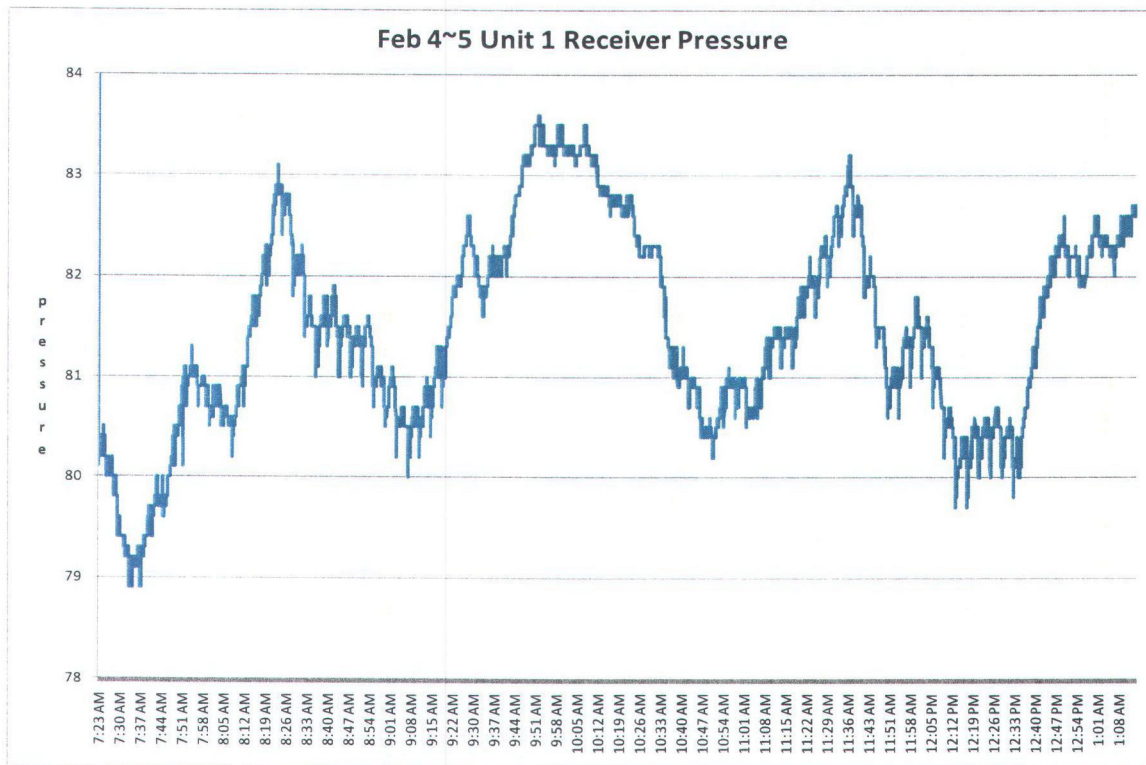


Figure 10. Unit #1 Header Pressure

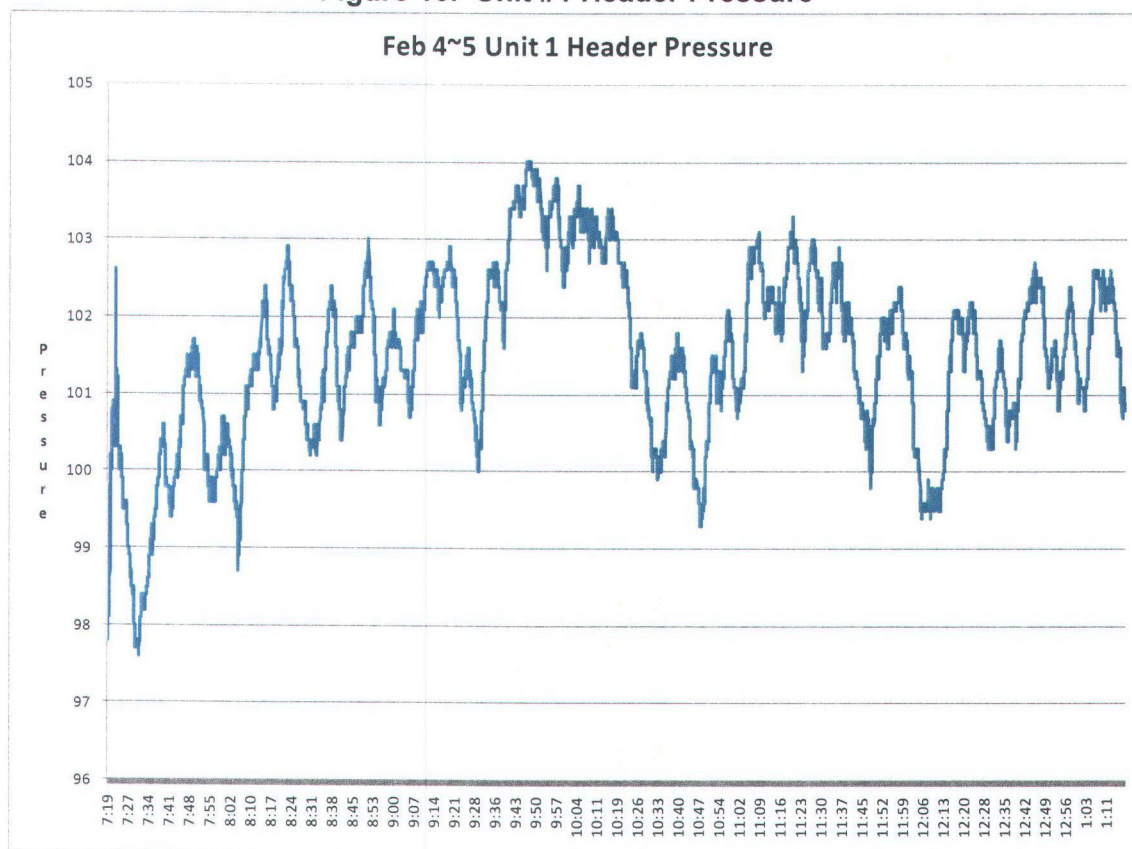


Figure 11. Bag House Pressure Test

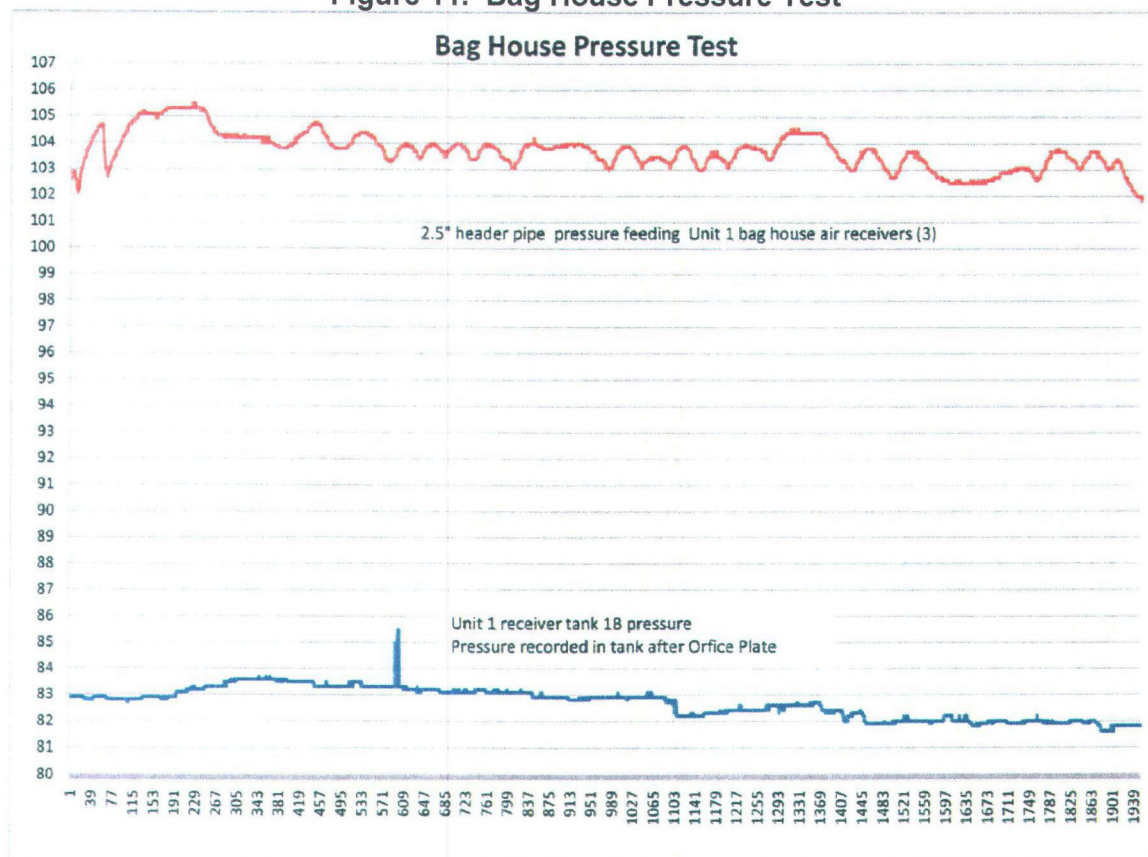


Figure 12. Unit #2 Bag House Cleaning Cycles

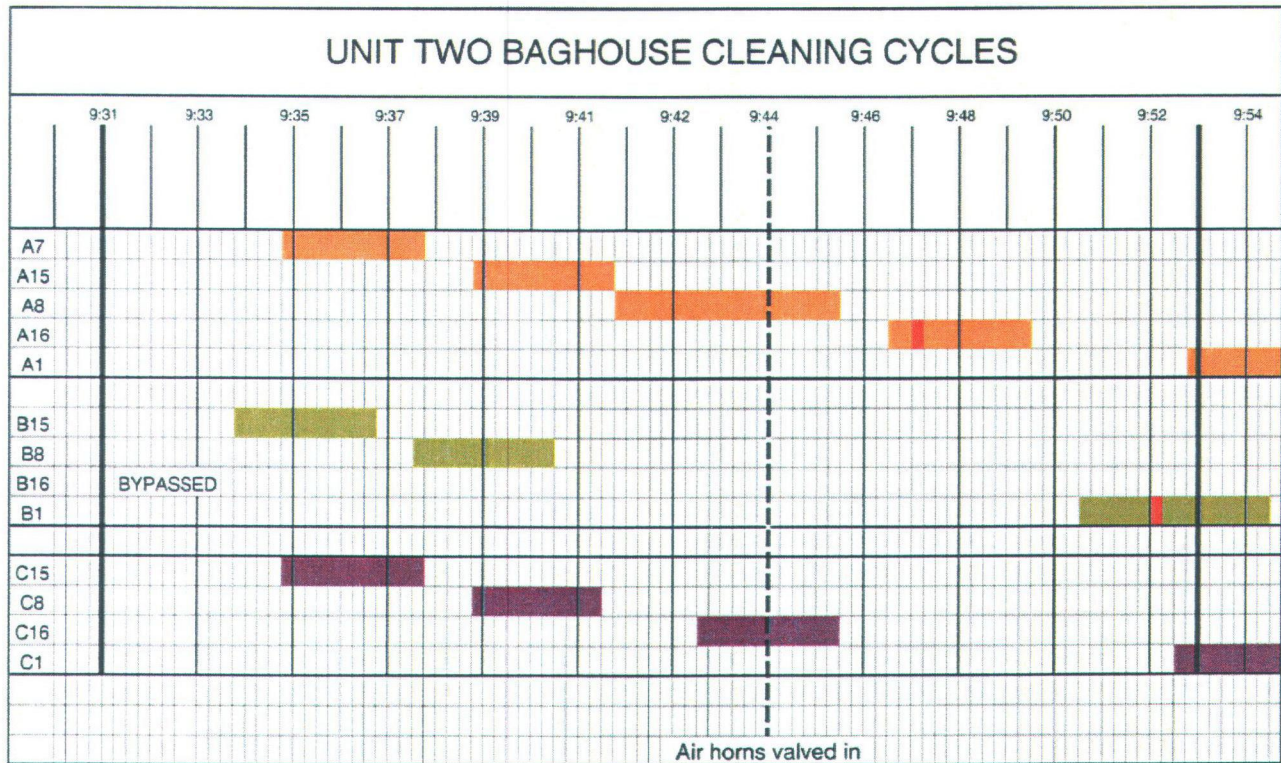
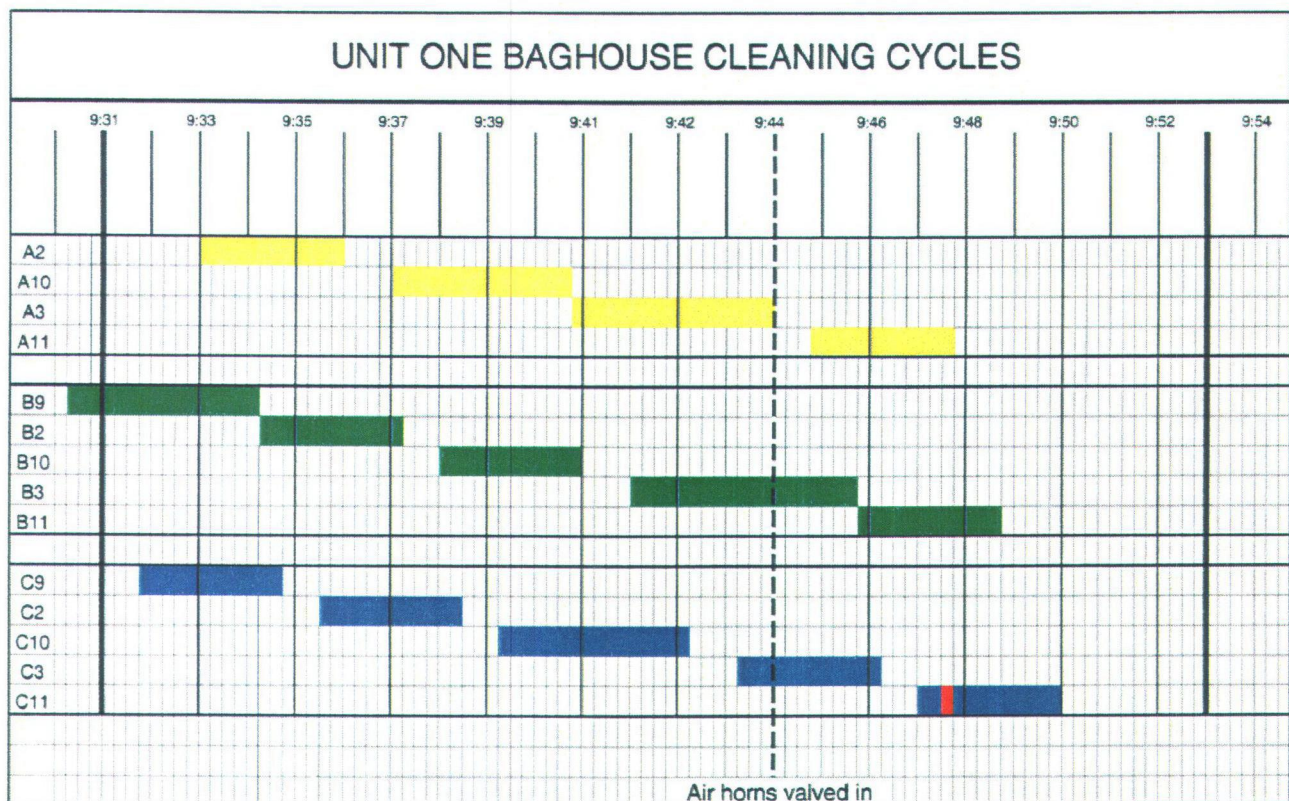


Figure 13. Unit #1 Bag House Cleaning Cycles



## Summary

Establishing the baseline for air demand flow and pressure for the Intermountain Power Service Corp. Electric Generating Station in Delta, Utah.

Measured demand as running today at 119 psig average pressure after the dryer discharge, reference the preceding line diagram and schematic:

|  |            |
|--|------------|
| Unit service flow average                      | 1,365 scfm |
| Non-essential induce and outside air (average) | 1,165 scfm |
| Units #1 and #2 Bag Houses average             | 2,864 scfm |
| Heatless dryer purge air average               | 1,335 scfm |
| Total (average)                                | 6,729 scfm |

Required system entry pressure is 100 psig.

Historical data shows at this similar operation. Last year, three compressors and four dryers ran 73% of the time and four compressors and four dryers ran the remainder of the time.

Controls – today's units operate with no turndown.

## Operating Profile

The following chart titled "4 Compressors in Service" was supplied by plant personnel, which indicates that from March 9, 2007 to February 22, 2008, four units were on 27% of the time.

|   |              |
|---|--------------|
| Four units operating hours (8,760 x .27)    | 2,365 hrs/yr |
| Three units operating hours (8,760 - 2,365) | 6,395 hrs/yr |

## Current System Baseline

Tables 1 and 2 reflect the energy and economic performance of the current air system.

**Table 1. Key Air System Characteristics – Current System\***

| Measure   | Scenario A<br>3 Compressors<br>and 4 Dryers on<br>All Shifts | Scenario B<br>4 Compressors<br>and 4 Dryers on<br>All Shifts | Total               |
|---|--|--|---------------------|
| Average System Flow   | 6,729 cfm  | 7,259 cfm  | NA                  |
| Avg Compressor Discharge Pressure   | 125 psig   | 125 psig   | NA                  |
| Average System Pressure   | 105 psig   | 102 psig   | NA                  |
| Input Electric Power  | 1,566.36 kW  | 2,088.48 kW  | NA                  |
| Operating Hours of Air System   | 6,395 hrs  | 2,365 hrs  | 8,760 hrs           |
| Specific Power  | 4.29 scfm/kW   | 3.4 7cfm/kW  | 3.96 scfm/kW        |
| Electric Cost for Air /Unit of Flow   | \$74.43 /scfm year   | \$34.02 /scfm year   | \$108.45 /scfm year |
| Ann'l Elec Cost for Compressed Air<br>(Main Year-round Production)  | \$500,844 /year  | \$246,963 /year  | \$747,807 /year     |
| Summer Air Horn Use:<br>(10 air horns x 124 scfm each x 1 kW per<br>scfm x 2190 hours in summer season x 5¢<br>per kWh) | \$27,156 /year   |  |                     |
| TOTAL ANNUAL COST   | \$775,071 /year  |  |                     |

\*Based on a blended electric rate of \$0.05 per kWh, 8,760 hours/year.

## Comments

The current compressors have local modulation control and are operating with no measurable turndown.

**Table 2. Compressor Use Profile – Current System**

| Unit #  | Compressor: Manufacturer/Model | Full Load   |                 | Actual Elec Demand |           | Actual Air Flow |             |
|---|--------------------------------|-------------|-----------------|--------------------|-----------|-----------------|-------------|
|   |                                | Demand (kW) | Air Flow (scfm) | % of Full kW       | Actual kW | % of Full Flow  | Actual scfm |
| First Shift: Operating at 125 psig discharge pressure for 6395 hours  |                                |             |                 |                    |           |                 |             |
| 1   | 3 Stage Elliott                | 522.12      | 2245            | 100                | 522.12    | 100             | 2245        |
| 2   | 3 Stage Elliott                | 522.12      | 2245            | 100                | 522.12    | 100             | 2245        |
| 3   | 3 Stage Elliott                | 522.12      | 2245            | 100                | 522.12    | 99              | 2239        |
| 4   | 3 Stage Elliott                | 522.12      | 2245            | OFF                |           |                 |             |
| TOTAL (Actual):   |                                |             |                 | 1566.36 kW         |           | 6729 scfm       |             |
| Second Shift: Operating at 125 psig discharge pressure and 2365 hours |                                |             |                 |                    |           |                 |             |
| 1   | 3 Stage Elliott                | 522.12      | 2245            | 100                | 522.12    | 100             | 2245        |
| 2   | 3 Stage Elliott                | 522.12      | 2245            | 100                | 522.12    | 100             | 2245        |
| 3   | 3 Stage Elliott                | 522.12      | 2245            | 100                | 522.12    | 99              | 2239        |
| 4   | 3 Stage Elliott                | 522.12      | 2245            | 100                | 522.12    | 23              | 524         |
| TOTAL (Actual):   |                                |             |                 | 2088.48 kW         |           | 7259 scfm       |             |

### Current System Summary

Annual plant electric costs for air production, as running today, are \$775,071 per year. These estimates are based on a blended electric rate of \$0.05 /kWh.

The air system operates 8760 hours a year. The load profile or air demand of this system is relatively stable during all shifts

## 2.2 PROPOSED SYSTEM DESCRIPTION

The overall strategy for improving the air system centers on replacing the inefficient heatless dryers with blower purge combined with other air conservation programs to significantly reduce the amount of air demand. Reconfiguring piping where required to eliminate pressure losses.

### Proposed System Changes

The specific projects to improve the air system are described in Chapters 3 and 4 of the report. Figure 2 provides a schematic of the proposed system. The recommended projects include:

#### Efficiency Projects

- Install central air management control system with appropriate inlet guide vanes and other auxiliary equipment, such as the CEC quotation we reviewed , overall unit turndown should go up to 80% average turndown and up to 80% power. 3,118 scfm

#### Air Flow Reduction Projects (Total Reduction = 2794 scfm)

- Replace current heat of compression with blower purge units 1348 scfm
- Replace current aftercooler gravity drains with appropriate level activated (8) 110 scfm
- Reduce system pressure 19 psig to unit service air, non essential, inside, outside air 481 scfm
- Repair tagged leaks, continue program 855 scfm

#### Other Projects (Total Reduction = 208 kW)

- Run at least two units at/or near full turndown with the new air management control system. This is reflected in the proposed table by the effect of probable efficient turndown. 208 kW  
\$91,104/ yr
- Replace air-operated air horns when used during summer season (2190 hrs) 1240 scfm  
\$27,156

## Proposed System Impacts

Tables 3 and 4 reflect the impact the proposed projects are expected to have on air system performance and operating costs of the current system reported in earlier Tables 1 and 2.

There are two categories of savings: savings reflected in comparing the compressor operating costs in the current system (Table 1) and the proposed system (Table 3), and additional savings not directly associated with operating the compressors, such as adding cycling refrigerated dryers.

**Table 3. Key Air System Characteristics – Proposed System\***

| Measure   | Scenario A<br>2 Compressors<br>and 2 Dryers on<br>All Shifts   | Scenario B<br>2 Compressors<br>and 2 Dryers on<br>All Shifts | Total             |
|---|--|--|-------------------|
| Average System Flow   | 3935 scfm  | 4465 scfm  | NA                |
| Avg Compressor Discharge Pressure   | 110 psig   | 110 psig   | NA                |
| Average System Pressure   | 100 psig   | 100 psig   | NA                |
| Input Electric Power  | 856 kW   | 972 kW   | NA                |
| Operating Hours of Air System   | 6395 hrs   | 2365 hrs   | 8760 hrs          |
| Specific Power  | 4.60 scfm/kW   | 4.59 scfm/kW   | NA                |
| Electric Cost for Air /Unit of Flow   | \$69.56 /scfm year   | \$25.74 /scfm year   | \$95.30 /cfm year |
| Ann'l Elec Cost for Compressed Air  | \$273,706 /year  | \$114,939 /year  | \$388,645 /year   |
| Summer Air Horn Use:<br>(no air horns x 124 scfm each x 1 kW per<br>scfm x 2190 hours in summer season x 5¢<br>per kWh) | (Negligible additional electric use with electric-operated air<br>horns – less than \$1,000 per year)i |  |                   |
| TOTAL ANNUAL COST   | \$388,645 /year  |  |                   |

\*Based on a blended electric rate of \$0.05 per kWh, 8760 hours/year.

**Table 4. Compressor Use Profile – Proposed System**

| Unit #  | Compressor: Manufacturer/Model | Full Load   |                 | Actual Elec Demand |           | Actual Air Flow |             |
|---|--------------------------------|-------------|-----------------|--------------------|-----------|-----------------|-------------|
|   |                                | Demand (kW) | Air Flow (scfm) | % of Full kW       | Actual kW | % of Full Flow  | Actual scfm |
| First Shift: Operating at 110 psig discharge pressure for 6395 hours  |                                |             |                 |                    |           |                 |             |
| 1   | 3 Stage Elliott                | 522.12      | 2402            | 82%                | 428       | 82%             | 1968        |
| 2   | 3 Stage Elliott                | 522.12      | 2402            | 82%                | 428       | 82%             | 1967        |
| 3   | 3 Stage Elliott                | 522.12      | 2402            | OFF                |           |                 |             |
| 4   | 3 Stage Elliott                | 522.12      | 2402            | OFF                |           |                 |             |
| TOTAL (Actual):   |                                |             |                 | 856 kW             |           | 3935 scfm       |             |
| Second Shift: Operating at 110 psig discharge pressure and 2565 hours |                                |             |                 |                    |           |                 |             |
| 1   | 3 Stage Elliott                | 522.12      | 2402            | 93%                | 486       | 93%             | 2233        |
| 2   | 3 Stage Elliott                | 522.12      | 2402            | 93%                | 486       | 93%             | 2232        |
| 3   | 3 Stage Elliott                | 522.12      | 2402            | OFF                |           |                 |             |
| 4   | 3 Stage Elliott                | 522.12      | 2402            | OFF                |           |                 |             |
| TOTAL (Actual):   |                                |             |                 | 9724 kW            |           | 4465 scfm       |             |

### Proposed System Summary

The savings potential of the projects related to operating the compressors and a project of \$1,212 per year to operate the new dryers total \$385,214. Costs for implementing these projects still need to be quoted, but the total cost is expected to be less than \$770K or a two-year payback.

Some of the key parameters characterizing the current and proposed systems and the associated savings projects are provided below.

| SYSTEM COMPARISON                  | CURRENT SYSTEM    | PROPOSED SYSTEM                |
|------------------------------------|-------------------|--------------------------------|
| Average Flow (Scenario A / B)      | 6729 scfm 7259    | 3935 scfm 4465                 |
| Avg Compressor Discharge Pressure  | 125 psig 125      | 110 psig 110                   |
| Average System Pressure            | 105 psig 105      | 100 psig 100                   |
| Electric Cost per Cfm              | \$108.45 /scfm/yr | \$95.30 /scfm/yr               |
| Annual Electric Cost:              |                   |                                |
| Compressor Operation               | \$775,071         | \$388,645                      |
| Other Air Equipment                | \$*               | \$1,212 (new dryers)           |
| Total Annual Electric Cost         | \$775,071         | \$389,857                      |
| <b>OVERALL PROJECT EVALUATION:</b> | <b>SAVINGS</b>    | <b>COSTS</b>                   |
| Total                              | \$385,714         | < \$770k<br>(< 2-year payback) |

\*Note: The current dryers are heatless and have no direct energy cost but use 1348 scfm in purge air at \$84.53 / scfm/yr which equals \$113,946 per year. Replacing these is reflected in the overall compressor operating cost. The replacement heated blower purge dryers have a projected operating cost of \$1,212 per year with two dryers running and the effective purge control.

## 2.3 PROJECT EVALUATION METHODOLOGY

The specific supply-side and demand-side projects that form the basis of the new proposed compressed air system are described and evaluated in Chapters 3 and 4. In order to provide a reasonable value of the savings associated with each project, a methodology is used to allocate the total system savings among the individual projects. Such a methodology is motivated, in part, by seeking to avoid any potential double counting in savings estimates – a common mistake in many compressed air assessments.

The methodology is based on determining parameters for the “\$ per psig saved” for pressure reduction projects, “\$ per cfm saved” for flow reduction projects, and the “\$ saved” for compressor efficiency and reconfiguration projects. Although any allocation approach can result in the savings parameter being set too high for one type of project (e.g., pressure reduction projects) and, correspondingly, too low for a second project type (e.g., flow reduction projects), summing the total savings for all the individual projects will match the total system cost improvement derived in Section 2.2.

In any case, it is always recommended that the entire set of recommended projects be implemented, because many of the projects are interactive in nature. Leaving out a single project could eliminate the effectiveness of the remaining projects that are implemented. Proposed air systems can also improve air quality, reduce maintenance costs, extend equipment life, reduce water use, improve environmental compliance, and reduce rental costs. For example, associated maintenance and other costs can often enhance project savings by 30% of the identified electric cost savings. If important to the assessment, these other savings can be tracked in addition to the electric cost reductions derived in Section 2.2.

Most of the overall program savings is simply the difference between the operating costs of the current compressors (\$775,071 -- Table 1) and the proposed compressors (\$388,645 -- Table 3) or \$386,426. This figure needs to be decreased by \$1,212 to \$385,214 to reflect the direct electric use by the new drying system (Project #3).SED

Estimated savings from the supply-side projects (Projects #1 and #2) are derived in Section 3.1 and total \$123,094 (\$31,990 + \$91,104).

This leaves \$263,332 (\$386,426 - \$123,094) to be allocated among the air flow reduction projects (Projects #3 - #9). Of this amount, \$27,156 is associated with the air horn project (Project #8) described in Section 4.5.1. The remaining of \$236,176 (\$263,332 - \$27,156) is allocated among the air flow reduction projects, which are saving 2794 scfm at a calculated value of \$84.53 per acfm.

## CHAPTER 3. SUPPLY-SIDE SYSTEM REVIEW

### 3.1 PRIMARY AIR COMPRESSOR SUPPLY

The primary air compressors are early 1980s technology, 3-stage, 100-psig class Elliott centrifugal compressors. There have been at least two generations of significant technological performance improvement since these units were produced. The currently available enhanced performance of this class of compressors has been a product of much more precise manufacturing capability and the ability of design engineers to improve upon previous design utilizing this more precise manufacturing.

New centrifugal compressors of this class will have a 10 to 20% better basic specific power. Intermountain Power has a proposal to add some specific product upgrades and accessories that will have a very significant positive impact on performance:

- Inlet guide vanes to allow full available turndown at a very favorable specific power.
- Target pressure controlled, full networking central air management system.

Installation and upgrading these units with proper auxiliary and accessory equipment will offset some of this inefficiency. Overall, the units appear to be well maintained and in generally good working order, except as noted.

**Table 5. Comparison of Current and Proposed Compressor Ratings**

| Manufacturer                                | Elliott (#1) | Elliott (#2) | Elliott (#3) | Elliott (#4) |
|---|--------------|--------------|--------------|--------------|
| Model                                       | 310DA3       | 310DA3       | D10DA3       | 310DA3       |
| Unit Type                                   | 3-stage Cent | 3-stage Cent | 3-stage Cent | 3-stage Cent |
| Type of Cooling                             | Water        | Water        | Water        | Water        |
| Full Load Nominal Published BHP             | 700          | 700          | 700          | 700          |
| Full Load Horsepower (bhp) actual           | 660          | 660          | 660          | 660          |
| Full Load Motor Efficiency (.me)/calc       | .943/.90     | .943/.90     | .943/90      | .943/.90     |
| Full Load Pressure (psig)                   | 108          | 108          | 108          | 108          |
| Full Load Flow (icfm)                       | 3,100        | 3,100        | 3,100        | 3,100        |
| Full Load Flow (scfm)                       | 2,245        | 2,245        | 2,245        | 2,245        |
| Full Load Nominal Set Point (psig)          | 108          | 108          | 108          | 108          |
| Type of Capacity Control                    | IBV/BOV      | IBV/BOV      | IBV/BOV      | IBV/BOV      |
| Pressure Control Band                       | 125-135      | 125-135      | 125-135      | 125-135      |
| Turn Down % (estimated avg)                 | 11%          | 11%          | 11%          | 11%          |
| Turn Down Air Flow (scfm)*                  | 1,998        | 1,998        | 1,998        | 1,998        |
| Full Load (input) kW @ 108 psig: Calculated | 522.12       | 522.12       | 522.12       | 522.12       |
| Turn Down kW: 11% (5% power)                | 496.02       | 496.02       | 496.02       | 496.02       |
| Idle kW (estimated)                         | 183          | 183          | 183          | 183          |
| Full Load Specific Power (scfm/kW)          | 4.834        | 4.834        | 4.834        | 4,834        |
| Annual Electric Cost (\$/scfm)*             | \$121.05     | \$121.05     | \$121.05     | \$121.05     |

\* Based on blended electric rates of \$0.05 per kWh and operation of 8,760 hours per year.

\*\*Plant uses 2,245 scfm each for capacity.

acfm to scfm multiplier = x .7315

$$100 \times \frac{(12.2 \text{ psia} - .9492 \text{ psia})}{14.5 \text{ psia}} \times \frac{528^\circ\text{F}}{460 + 100^\circ\text{F}} = \frac{528}{560}$$

$$100 \times \frac{(11.25)}{14.5} \times \frac{528}{560} = .7315$$

Scfm calculated:

68°F / 14.5 psia / 0% RH

Operating Condition:

Ambient Temp 100°F used / 20% RH

$$\text{kW} = \frac{\text{amp} \times \text{volts} \times 1.732 \times \text{PF}}{1,000}$$

$$522.12 = \frac{(\text{amps}) 6600 \times 1.732 \times .90}{1,000}$$

Full Load:

Estimated amps at 6600 volts = 50.75 kW

Estimated full turndown amps at 11% TD = 48.21 amps

- ☒ **RECOMMENDED PROJECT (#1)** – Reduce compressor discharge pressure from 125 psig to 110 psig after the piping has been reconfigured.

This will increase the design air flow 7%, but operating at about the same power, and extend the turndown range. Standard 125 psig flow – 2,245 scfm plus 7% extra air from each operating unit about the same energy use.

Each compressor can now deliver 2,402 scfm each ( $2,245 \times 1.07$ ) for a gain of 157 scfm each or 628 scfm for four operating units. Because the proposed system is expected to run with just two units turned on, the estimated actual air flow increase with two units on is 314 scfm.

|   |                   |
|---|-------------------|
| Net additional air (2 units @ 157 scfm increase each)   | 314 scfm          |
| Value of additional air -- average cost of air between current system (\$ 108.45 /scfm per year in Table 1) and the proposed system (\$95.30 /scfm per year in Table 3) | \$101.88 /scfm yr |
| Total value of additional air   | \$31,874          |
| Project costs – included in Projects #2 and #6  |                   |

### 3.2 COMPRESSOR CAPACITY CONTROL

The two most effective ways to run air compressors are at “Full Load” and “Off.”

Capacity controls are methods of restricting the output air flow delivered to the system while the unit is running. This is always a compromise and is never as efficient as full load on a specific power (cfm/hp) basis. For details on unloading, see the MISCELLANEOUS SECTION in the back of this report.

#### Centrifugal Controls

The two most common controls used for centrifugal compressors are **modulation** and **blow off**. Modulation is relatively efficient at very high loads, but will not work much below 70-75% load. The four units at Intermountain Power Delta have 11% design turndown on three units and 15% on the other unit. After “modulation” or “turn-down”, the compressor will then just “blow off” excess air. The basic power draw at the blow off point will stay the same regardless of the load. The actual operating turndown of the units as installed is really very low – apparently less than 5%. With the current type of inlet butterfly valve operators find it difficult, if not impossible, to securely avoid “surge” due to the active turbulence as the valve closes. The net result is little, if any, actual turndown. More importantly, IGVs make it routine to be able to take advantage of the full turndown. There are many times because centrifugals are a mass flow-type compressor when atmospheric conditions will allow greater than designed turndown.

A modern, well-applied electronic air management system, combined with effective inlet guide vanes will greatly enhance the operating efficiencies of these particular compressors.

For more information on inlet guide vanes, see the MISCELLANEOUS SECTION – article reprint from Plant Services entitled “Control the Air.”

Installing a management control system will allow the average of up to 20% turndown at 80% inlet power versus no turndown and 100% of power currently. This is a minimum savings of 104 kW per unit at part load or an average of two units at most times or 208 kW.

There are many other reasons to implement a professional compressed air management system including:

- Efficient, effective, and timely response.
- Improved and lower cost predictive and preventative maintenance.
- Allows proper performance checks
- Enhanced reliability and documentation of unscheduled downtime.

However, in all likelihood, this system will also probably reduce the overall energy operating electric energy cost at the compressor motor input.

- ☒ **RECOMMENDED PROJECT (#2)** – Install a new, modern electronic compressed air management control system combined with individual unit inlet guide vanes, replacing the current inlet butterfly valve system.

|  |          |
|--|----------|
| Net projected average kW reduction (running projected two units) | 208 kW   |
| Annual electrical energy cost (\$0.05 kW @ 8,760 hrs/year)       | \$91,104 |
| Estimated cost of project  | TBD      |

### 3.3 AIR TREATMENT AND AIR QUALITY

#### 3.3.1 Dryers

Desiccant dryer equipment removes moisture vapor by “adsorbing” it to desiccant beads (see MISCELLANEOUS SECTION). These dryers can consistently deliver a pressure dew point to -40°F or lower, which means they will remove more water vapor than refrigeration units. They regenerate the wet tower, while the other tower is drying. This requires the use of some type of heat and dry air to “sweep” or “purge” the exchanged moisture out.

The most common type of desiccant dryer is a twin tower, regenerative, desiccant dryer. These are most capable of delivering a consistent nominal -40° pressure dew point at rated scfm flow and purge when:

- Air is delivered to the dryer at less than 100°F
- Air is delivered to the dryer at no less than 100 psig
- Ambient air temperature is no more than 100°F
- The condensate is driven out of the aftercooler, pre-filter, and dryer is immediately removed from the system and is not allowed to re-entrain or build up
- No liquid water enters the dryer
- The dryer is not overloaded in volume (scfm)
- Air is not re-contaminated by moisture “wicking” past air leaks in the system
- Inlet air at 130°F or more will not be dried at all.

#### Water or Oil Carryover in System

Water (condensate) and oil carryover problems in the current air system are not significant when the dryers are working. The current dryers are almost 25 years old and utilize some older-style valves and controls. Currently, they have not shown a high degree of reliability according to plant personnel. Any problems can usually be expected to increase in magnitude during more humid months. The correct way to eliminate water and oil in the air system is to clean and dry the air immediately after it is produced in the compressor room. Then clean dry air can be stored in a separate air receiver and can flow to the system, as required. Some guidelines include:

1. Generally, it is best to eliminate water/oil at the air source before they enter air system.
2. Water vapor, when condensed to liquid in the drying process, must be removed immediately or it can recontaminate compressed air by evaporation and overflow.
3. Every 20°F increase in temperature will almost double the “moisture load” that air will hold. Compressed air dryers are usually capacity rated at 100°F and 100 psig inlet air conditions. At 120°F and 100 psig, the dryer’s capacity rating is reduced by 50%.
4. Putting dry/oil-free air into the system 90% of the time and then allowing wet/oily air to enter sporadically 10% of the time will, in reality, make the system wet all the time.

The water and/or oil will fall out in the piping system and continue to re-entrain and contaminate and/or collect in the “low spots” of the system. This will cause recontamination as liquid is pulled into the flowing compressed air system. Bypassing the dryer with “part of the air” (controlled by the bypass valve) will almost always end up with “wet air”! A wet system could take many months of continued flow of clean dry air in order to “clean up.”

5. It is best to identify required pressure dew point and meet it. Performance should be monitored closely, if critical. Intermountain Power Delta basically requires instrument-quality compressed air able to handle control valves and controls under winter conditions. It should be pointed out that extremely cold air holds insignificant volumes of water vapor. Many winter problems with “freeze up” come from condensate build up in low spots during more humid times, which either re-entrains into the system or actually freezes in a critical spot.

### Current Drying System

Key features of the plant’s current dryers are displayed in Table 6, along with the key features of proposed dryers recommended in this report.

**Table 6. Comparison of Current and Proposed Dryers**

| Manufacturer                       | Current        |                  | Proposed            |                     |
|------------------------------------|----------------|------------------|---------------------|---------------------|
|                                    | Pall Trinity   | Total of 4 Units | Desiccant           | Total 4 Units       |
| Model                              | Desiccant      | 4 ??             | Desiccant           | --                  |
| Unit Type                          | Heatless       | Heatless         | Heated blower purge | Heated blower purge |
| Rated Flow @ 100°F/100 psig        | 2,242          |                  | 2,242               | --                  |
| Purge: scfm (CompAir)              | 337            | 1,348            | NA                  | NA                  |
| Full Load Heater kW                | NA             | NA               | 36x.75=27 avg kW    | 108 avg kW          |
| Full Load Blower hp/kW             | NA             | NA               | 15/12.5             | 50 kW               |
| Total kW                           | NA             | NA               | 39.5                | 158                 |
| % Load w/ Dew Point Demand Control | 100%           | 100%             | 35%*                | 35%                 |
| Net Purge                          | 337            | 1,348            | 13.83               | 55.32               |
| Total Annual Cost (\$)             | \$28,487 /year | \$118,948 /year  | \$605,765 /year     | \$2,423 /year       |

Based on blended electric rates of \$0.05 per kWh and operation of 8,760 hours per year.

\*Proposed dryers equipped with dewpoint demand controller.

- ☑ **RECOMMENDED PROJECT (#3)** – Replace four current heatless dryers with blower purge-type of similar rating with automatic dewpoint demand controllers. Eliminate 1,348 current purge blow off.

|   |               |
|---|---------------|
| Total annual operating cost of current dryer (1,348 scfm) [Table 6]           | \$113,946 /yr |
| Total annual operating cost of proposed dryer (for two compressors) [Table 6] | \$1,212 /yr   |
| Total annual electrical energy cost savings                                   | \$112,734 /yr |

Regeneration is accomplished by external heater and blower purge flows and the proposed dryer will be equipped with appropriate purge controls.

### Air Suitable for Breathing

If the application calls for purified air for facemasks, hoods, helmets, and other supplied-air breathing apparatus, you may need a breathing air system. There are complete utilized, purification systems designed to remove excessive moisture, solid particulates (dust and dirt), oil and oil vapor, carbon monoxide, and other hydrocarbon vapors commonly found in ordinary compressed air. Air flows through a breathing air system, including a number of filter-purifying stages, and a catalyst to convert carbon monoxide to carbon dioxide. Various contaminants are removed at each stage until final "Grade D" level air is produced, which is air suitable for breathing under OSHA standards.

The following table outlines the basic OSHA and Canadian limitation on breathing air purity. Note the following regarding pressure dew point.

- Moisture dew point temperature – 10°F below ambient temperature (@ 1 atmospheric pressure)
- Dryness *not to exceed* (-)50°F at 1 atmospheric pressure.

Many operators believe that too dry air [below (+)10°F PDP] will make breathing uncomfortable due to excessive dryness. In any event, to meet the maximum limitation 10°F below ambient temperature at 1 atmospheric pressure means that a (+)40°F pressure dew point at 100 psig would be a (-)10°F dew point at 1 atmospheric pressure and would be acceptable to ambient or as low as 0°F.

### Grade D – Breathing Air\*

| Contaminant                                  | OSHA<br>(Resp. Prot 1910.134)                              | CSA   | Outlet Concentration at Rated Conditions  |
|--|--|---|---|
| Oxygen (%)                                   | 19.5 to 23.5   | 20 to 22  |   |
| Carbon Monoxide                              | 10 ppm   | 5 mL/m <sup>3</sup>   | 10 with a max inlet concentration of 135:5 with max inlet condition of 100  |
| Carbon Dioxide                               | 1000 ppm   | 500 mL/m <sup>3</sup>   | CO is converted to CO <sub>2</sub> ; although some CO <sub>2</sub> is adsorbed in the desiccant beds, high concentration of CO <sub>2</sub> at the compressor intake, in addition to the CO <sub>2</sub> produced by the purifier |
| Oil and Condensate Hydrocarbons              | 6.65 ppm<br>5 (mg/m <sup>3</sup> )                         | 1 (mgL/m <sup>3</sup> )   | 0   |
| Odor   | Lack of noticeable odor                                    | Free of any detectable odor   | None: purifier will remove gases contaminants normally removed by carbon  |
| Moisture Content<br>Dew Point<br>Temperature | 10°F (5.6°C) below ambient temperature (at 1 atm pressure) | 9°F (5°C) below the min temperature breathing air is exposed (at line pressure) | Does not exceed –50°F (–45.6°C) when purified @ 100 psig and reduced to 1 atm pressure  |

\*Contaminant and maximum allowable limit required by OSHA in the U.S. and Canada (OSHA 1910.134(i)(1) (ii) (Table 1).

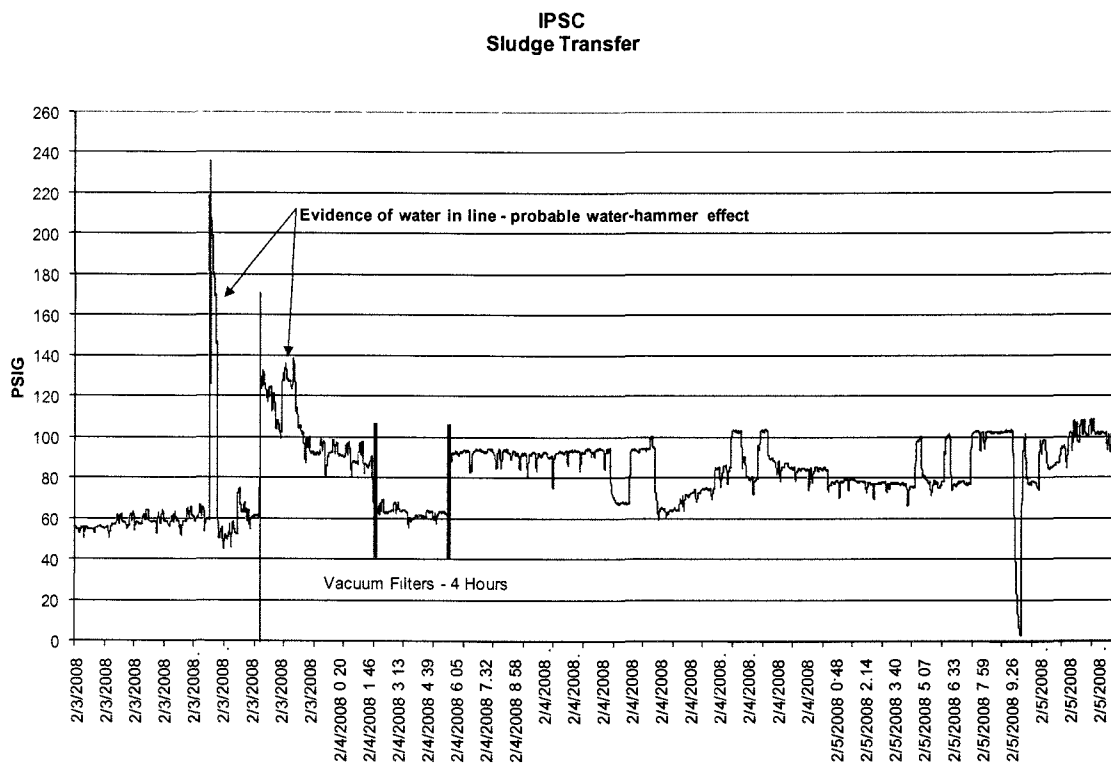
### Aftercoolers

Aftercoolers are water cooled and currently appear capable of delivering 100°F or lower temperature compressed air to the dryer. If the aftercooler is not performing correctly, then the dryers may be undersized and operating ineffectively, creating poor air quality.

Some key data on this:

- 20°F rise in inlet temperature above the rated temperature (normally 100°F) will double the moisture load on the dryer or reduce the dryer's capability by 50%
- Above 130°F, desiccant dryers will not dry at all. All water vapor goes downstream
- Liquid water entering a desiccant dryer will not be removed by adsorption to the beads – it will either go on through or react with the desiccant dust to plug the dryer and foul the element
- Liquid water going into refrigerated dryer water will suck off all the refrigeration due to the high latent heat. The water vapor may not be condensed and pass into the system as vapor to condense later on downstream
- Condensate/water will always run to gravity – for example, on the inside of the pipe
- Water vapor will always flow from a higher relative humidity to a lower, regardless of the air flow.

**Figure 14. Sludge Transfer**



## Summary

The complete air system is wet throughout all the headers and this water is giving the plant problems. There is rust and scale in the pipes, valves, etc. Water slugs or “water hammers” are spiking the pressure. Frozen water is blocking the lines and valves.

There are many open drains left cracked open to bleed water because of the problems.

The probable cause for this is the opening of the bypass valves around the dryers for whatever reasons. When wet air goes into the system, the system quickly becomes wet. We believe there are currently significant volumes of water stored in low spots, tanks, etc. This must be drained and/or evaporated out before the total system can become dry.

Once the new dryers are installed or the current dryers are rebuilt to a higher degree of reliability (which may not be possible due to age and obsolescence), due diligence to operation and maintenance combined with the success of Projects #6 and #7 should create an atmosphere and program to preclude this happening again.

Projects #6 and #7, if implemented successfully, will eliminate the excessive piping pressure loss and eliminate any reason to bypass the dryers.

Compressed air reduction projects should leave you with one swing/back-up compressor and one swing/back-up dryer, which should allow proper shutoffs and maintenance.

During the “drying up” process, the system may well create some possible significant accumulations of rust and scale for about 4 to 6 months, which will have to be addressed.

### 3.3.2 Condensate Drains and Handling

#### Background

Automatic drain traps come in three categories: Level-operated mechanically activated, dual-timer electronic, and level-operated electronic drains.

**Level-Operated Mechanically Activated Drains.** These drains do not waste air, but are prone to clogging and require continuing maintenance to assure operation. These drains work best in a “Power House Situation” where regular attention on an ongoing basis is part of the operation. Drain prices range from \$65.00 each to \$250.00 each.

**Dual-Timer Electronic Drains.** These drains use an electronic timer to control the number of times per hour it opens and the duration of the opening. The theory is that the times should be adjusted to be sure that the condensate drains fully and the open time without water is minimized, because it wastes compressed air. The reality is that the cycles often don’t get reset from the original factory settings. This results in condensate build-up during the summer and in getting set wide open and not closed down later during in cooler weather. When they fail “stuck open”, they blow at a full flow rate of about 100 cfm.

Consider, for example, that the usual “factory setting” is 10 minutes with a 20-second duration. Some 1500 scfm of compressed air will generate about 63 gallons of condensate a day in average weather or 2.63 gallons per hour. Each 10-minute cycle will have 0.44 gallons to discharge. This will blow through a ¼-inch valve at 100 psig in approximately 1.37 seconds. Compressed air will then blow for 18.63 seconds each cycle, 6 cycles a minute, which will total 111.78 seconds per hour of flow or 1.86 minutes per hour of flow. A 1/8-inch valve will pass about 100 cfm. The total flow will be  $100 \times 1.86 = 186$  cubic feet per hour, or  $186 \div 60$  minutes = 3.1 cu ft/min on average. This 3.5 cfm would translate into an energy cost of \$300 per year based on a typical air flow cost of \$100 per cfm year.

Depending on the type of discharge valve (whether it is solenoid-operated or motorized ball valve-operated and whether the timer is dual-type with test button or remote alarm), the valve prices range from \$89 to \$600 each.

**Level-Operated Electronic and Pneumatic Drains.** These drains come in a number of varieties, including ones that receive the signal to open from a condensate high level and the signal to close from a condensate low level. These drains waste no air and are the best selection from a power cost standpoint. Their reliability is usually many times greater than the level operated mechanical drains. Prices range from \$250 to \$850 for standard products (more for specials).

Be sure auto drains are set up to work effectively. Some guidelines include making sure all drains:

- Are not tied together to a common header
- Can be checked easily for operation
- Are properly “vented” to atmosphere, if necessary
- Are sized, piped “to” and “from” with the full capability to handle anticipated highest humidity weather loads

- Have a bypass bleed on the feed pipe
- Can be easily checked, if they are passing condensate.

Connect each drain's point (after-cooler, pre-filter, dryer, after-filter, receivers, and all risers) separately to individual level-activated electric or pneumatic drains to collect and direct the condensate to a proper handling point carry it in a large plastic vented line (4" or 6"). Be sure maintenance personnel can effectively and visually monitor the drain's action.

### Current Application

The configuration and performance of the condensate drains in the plant's compressor area do need to be modified.

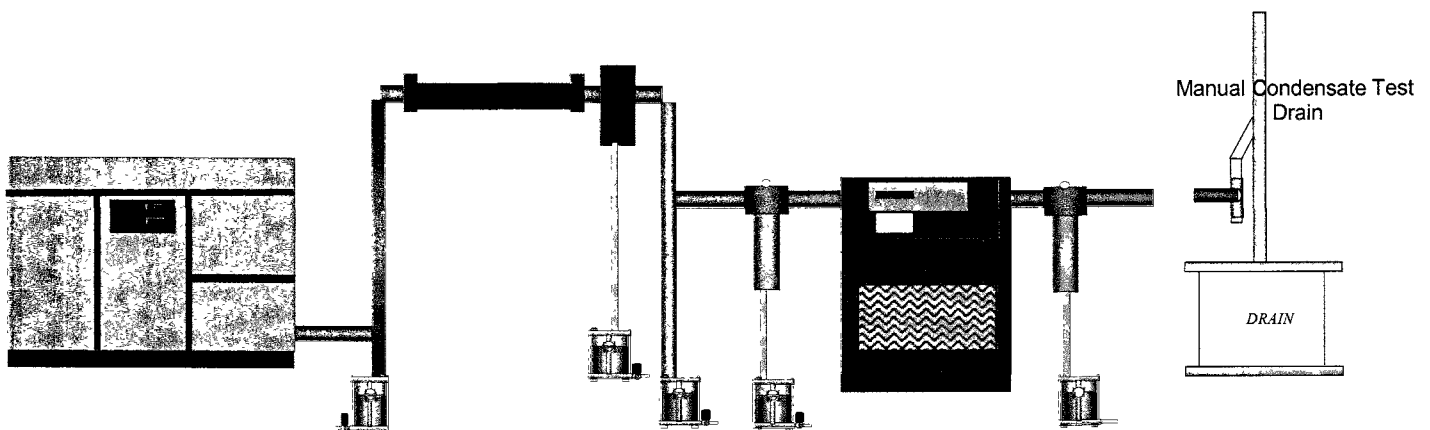
- ☑ **RECOMMENDED PROJECT (#4)** – The condensate drains from the aftercooler separators on the four primary compressors are valved to drain continuously. During our site visit, we noticed the aftercoolers from Unit 1D and 1C were both blowing significant amounts of compressed air continuously. The total lost air for all four compressors as observed is 110 cfm. We recommend the plant install two level-operated pneumatic-actuated drains rated at maximum handling capacity of 10-12 gph at each aftercooler dryer. Tie these together with a "Y" connection, 1" pipe to and from each drain to your current collector pipe. Also install one drain in the riser to the dryers.

|  |                     |
|--|---------------------|
| Total number of drains   | 8 pts/16 drains + 1 |
| Total compressed air saved                                       | 110 cfm             |
| Recoverable energy savings from air flow reduction [Section 2.3] | \$84.53 /cfm yr     |
| Total annual energy savings                                      | \$9,298 /yr         |

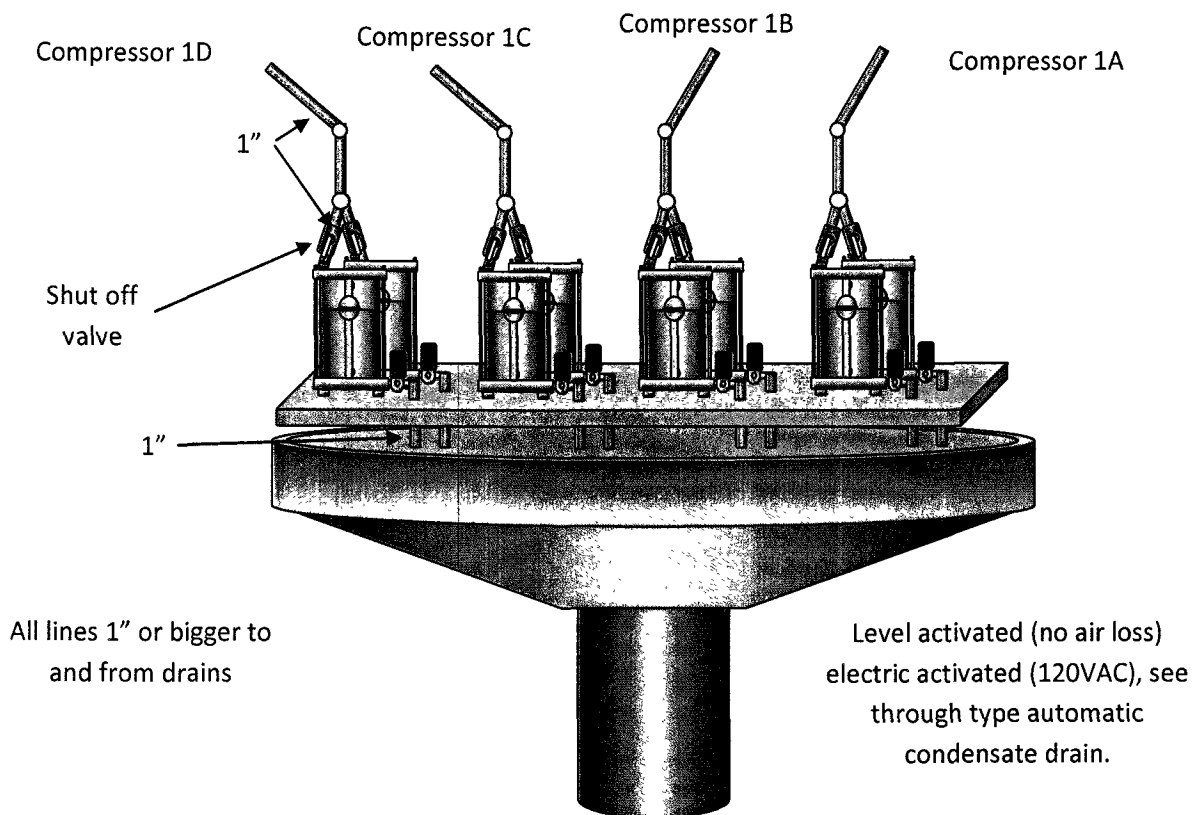
Cost per drain (materials and installation) – 17 drains for aftercooler/ 1 for riser) \$550 each

Cost of project (17 drains) \$9,350

**Figure 15. Level-Operated Drain on All Wet Side Risers and Drain Points**



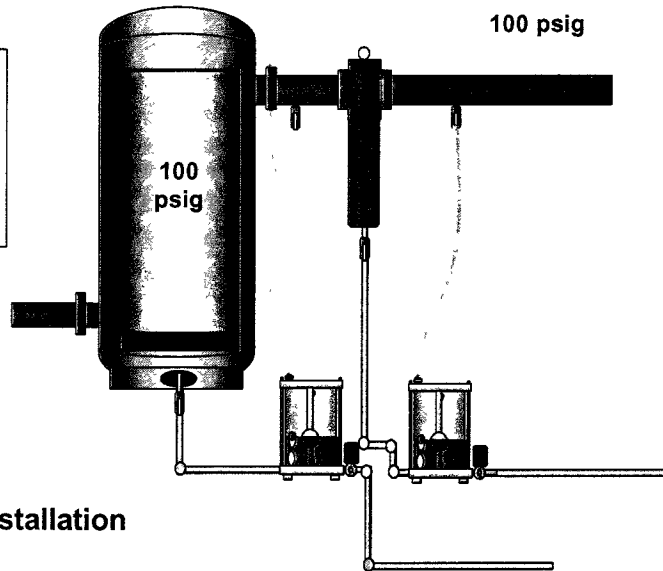
**Figure 16. Compressor Aftercooler Drain Suggested Changes**



## Don't "Air Lock"

### Drain "Air Locked"

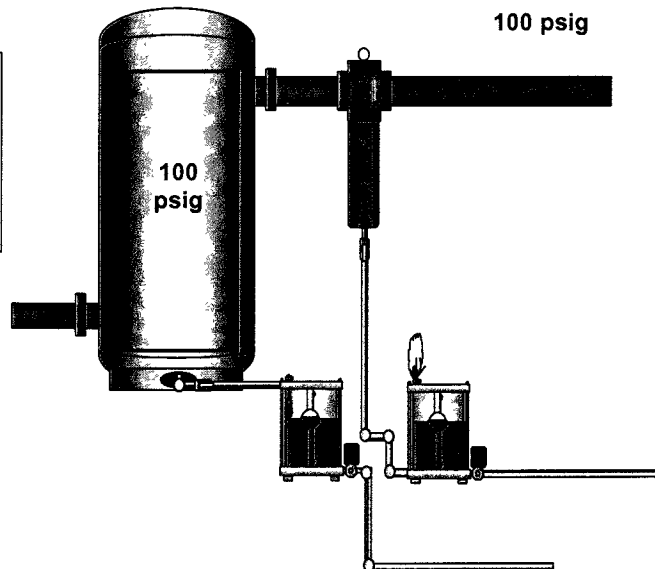
- Pressure equal – drains air lock – no additional condensate can come in
- Drains open – condensate stays in filter and/or air receiver



### Recommended Installation

### Recommendations to Avoid "Air Lock"

- Top feed into drain – no vent needed
- Vent line open to atmosphere with bottom feed



## Compressed Air Condensate Handling

Plant personnel state that the condensate goes to water treatment. If this is true, and if discharge condensate meets the requirements of the local water treatment facility, there is no problem. (Refer to the Article Reprint – "Do You Know Where Your Condensate Is?" in the MISCELLANEOUS SECTION.)

However, if the plant is discharging the condensate to a storm sewer or to some other ground water, the plant may be required to separate it by the local water treatment facility—Federal EPA minimum is 10 ppm. While there is no significant energy impact associated with most condensate handling projects, such projects can be critical to achieve environmental compliance and avoid environmental penalties. Cost for condensate-handling systems often fall in the \$3-10K range.

The compressed air condensate handling process at all plants should be reviewed to ensure environmental compliance.

## CHAPTER 4. DEMAND-SIDE SYSTEM REVIEW

### 4.1 BASIC SYSTEM HEADER AND PIPING

#### Background

It is the job of the main header system to deliver compressed air from the compressor area to all sectors of the plant, with little or no pressure loss. The header should be checked at appropriate points with a single test gauge. If there is a significant pressure drop anywhere, then corrective actions are likely needed. A pressure loss of no more than 1 to 3 psid is a reasonable target.

It is also desirable that the compressed air velocity in the main headers be kept below 20 fps to allow effective drop out of contaminants and to minimize pressure losses caused by excessive turbulence. The magnitude of the turbulence effect depends on the piping layout and pipe size.

Typical header projects include adding pipe, replacing pipe with larger diameters by adding angled connectors, and re-orienting or re-directing air flows.

#### Investigation of the Bag House Compressed Air Usage – Units #1 and #2 and the Overall Air Distribution Supply

Units #1 and #2 Bag House utilize reverse flow. Following specifications:

- Each unit has 3 casings
- Each casing has 16 compartments
- Each compartment has
  - 2 inlets
  - 2 outlets
  - 2 reverse air
  - purge air (fan air)
  - air horns
  - 2 vibrators.

Each casing has:

- 4 pre-casing cylinders (14"x66")
- 1 relief cylinder (5"x22")

Trended plant data has each compartment running 1 cycle every 24 hours and using about 2,646 cfm or 482 cfm per casing per Bag House.

- air horns (every third cycle) used to flutter the bags
- 2 air vibrators per component

The flow to these two Bag Houses is controlled by orifice plates. Originally, there were a total of six orifices installed on each of the sets of three air receivers feeding the compressed air to

each Bag House. These are .421" ID rated to flow 298 scfm each at 100 psig, 70°F inlet condition. Estimated total anticipated flow is 1,800 scfm.

Three years ago, larger orifice plates were installed on the 2" entry lines to the receivers for Unit #2. These new orifice plates have an ID of .621" and are still in use. These are rated to flow 630 scfm each at the same inlet conditions.

|                             |         |              |
|-----------------------------|---------|--------------|
| New total anticipated flow: | 3 @ 298 | 894          |
|                             | 3 @ 630 | <u>1,890</u> |
|                             |         | 2,784 scfm   |

#### Other Changes

These bag houses have had to handle more product with new coal supplies and the cycle time was increased, which would increase some of the air cylinder use.

Air horns (venturi air flow) were added to the process in order to increase the material handling capability with optimum bag life. Each air horn is rated to use 75 scfm at 75 psig (estimated 94 cfm @ 100 psig) entry pressure and moving 2,400 scfm with air horns to improve the performance of the reserve flow.

Three air horns operate every third cycle of five minutes and for 80-90 minutes per compartment. As the air horns go through the operating cycle, they are computer-controlled and blow for ten seconds (average flow of 12.5 scfm/rate of flow – 75 scfm @ 75 psig inlet pressure) every 4.5 minutes. They only operate in each compartment every third cycle. Their effect on the dynamics of the air system at this time appears to be insignificant.

#### Current Application

The current air distribution system was monitored throughout the audit process a 19 different points where calibrated pressure transducers with data loggers were mounted. The system sketch (pp.38-40) reflects the locations and average pressure readings during production as observed (for a complete set of trended downstream pressure profiles, see the PLANT SURVEY Section of this report).

Overall, the distribution system seems to be very adequate with a few notable exceptions as listed. Following are some of the significant areas where pressure loss is probably having a very negative impact on energy cost, reliability, and productivity:

Feeds from the main header to Units #1 and #2 Bag Houses. As described earlier in the baseline section, the air flow to the three receivers before the casings at each unit flow through restrictor orifice plates installed on the inlet to each receiver.

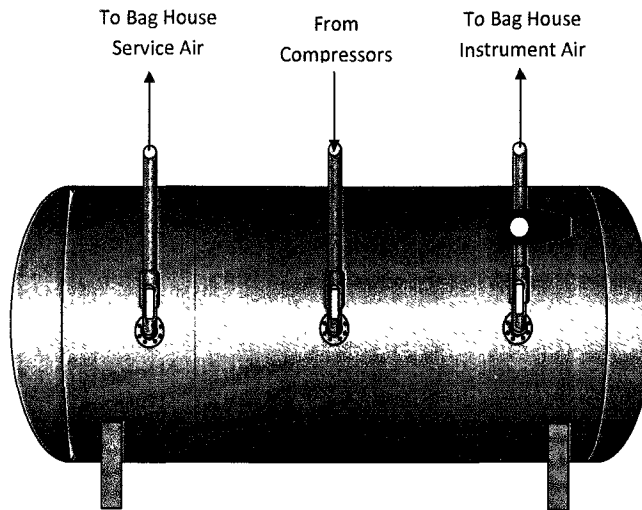
These are resulting in 20 psig loss to the Unit #1 Bag House (smallest orifice plate) and 10 psig to the Unit #2 Bag House (larger orifice plate).

Plant personnel state that the optimum operating pressure for the Bag Houses is 80 psig, which they cannot currently hold now in a continuing manner.

When all four dryers are on, the total pressure loss will probably make the sustainable entry pressure to the Bag Houses between 60 and 70 psig. The orifice plates were probably installed to sop an uncontrolled Bag House from pulling down other critical pressure areas.

- ☑ **RECOMMENDED PROJECT (#5)** – Remove orifice plates on the entry to each of the six air receivers. Install appropriately sized regulator recommend sized to handle 1,200 to 1,500 scfm with an 85 psig steady entry pressure and 80 psig delivered.

**Figure 17. Tank Recommended Changes**



Install these on the discharge line from each receiver to each casing. This will control the Bag House and protect other areas and allow better use of the stored air in the receivers.

There is a 16 psig loss in pressure from the dryer discharge to the main distribution header. This, of course, has a negative impact on all the following systems.

This is not an energy issue, but will allow better and more reliable operation.

- ☑ **RECOMMENDED PROJECT (#6)** – Reduce and control the main header entry pressure to 100 psig from 119 psig after re-piping and elimination of 16 psig loss from the dryer to the header.

Current flow of unit service air and non-essential indoor/outdoor air at 119 psig entry pressure at header (measured) = 2,530 scfm.

|  |            |
|--|------------|
| Projected air flow at 100 psig compressed air savings ( $2,530 \times .81$ ) | 2,049 scfm |
| Compressed air savings (part of re-piping project)                           | 481 scfm   |

For details, see the following schematics.

|   |                 |
|---|-----------------|
| Annual electrical energy cost per scfm per year | \$84.53/scfm/yr |
|---|-----------------|

Total scfm year saved (running two proposed units)  
Total recoverable annual electric energy savings

481 scfm  
\$40,659 /year

Figure 18. Typical Compressor Arrangement

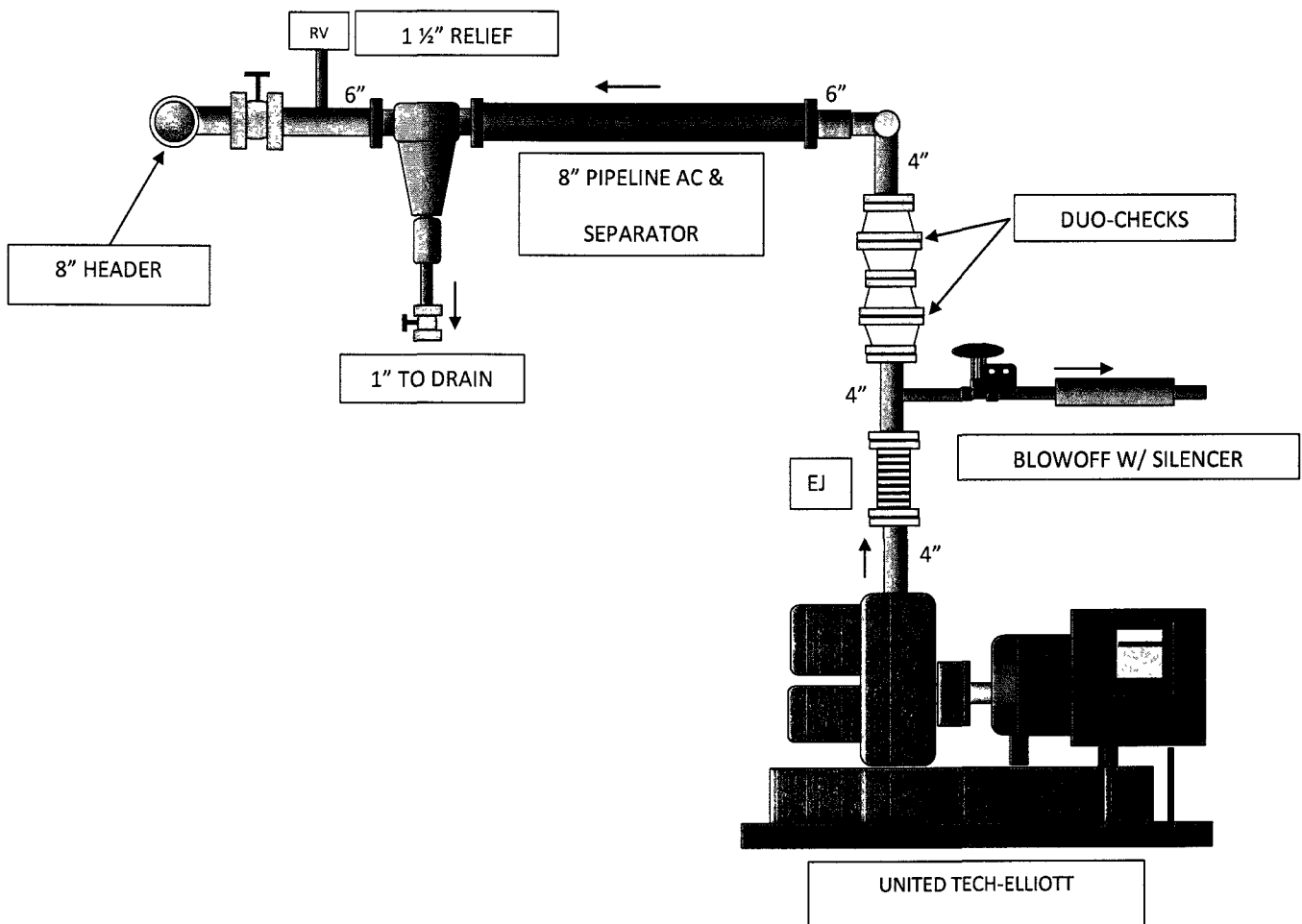
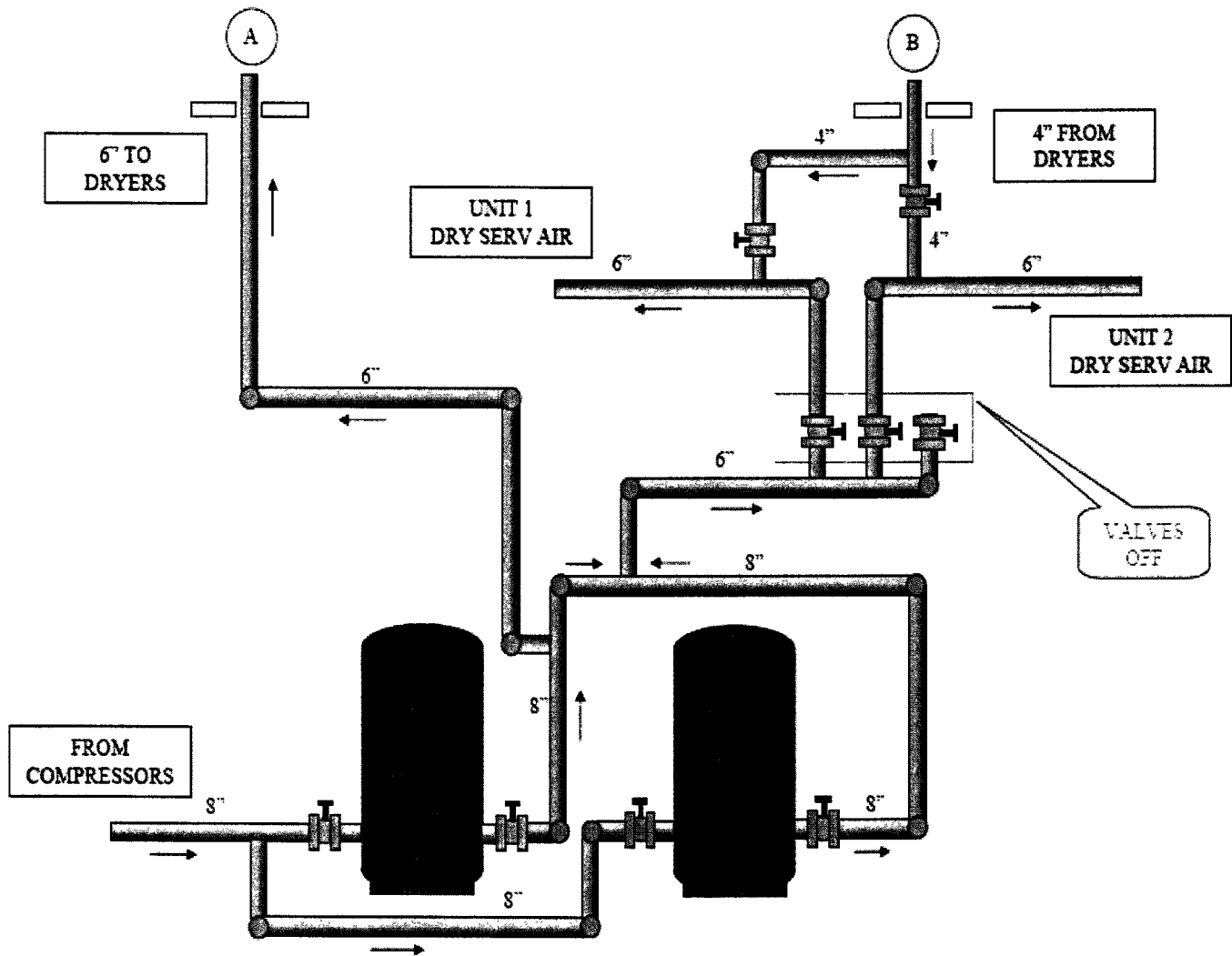
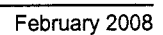


Figure 19. Compressor Floor Piping





**IP12\_006194**

Figure 21. Pressure Readings for Control Air Main Header

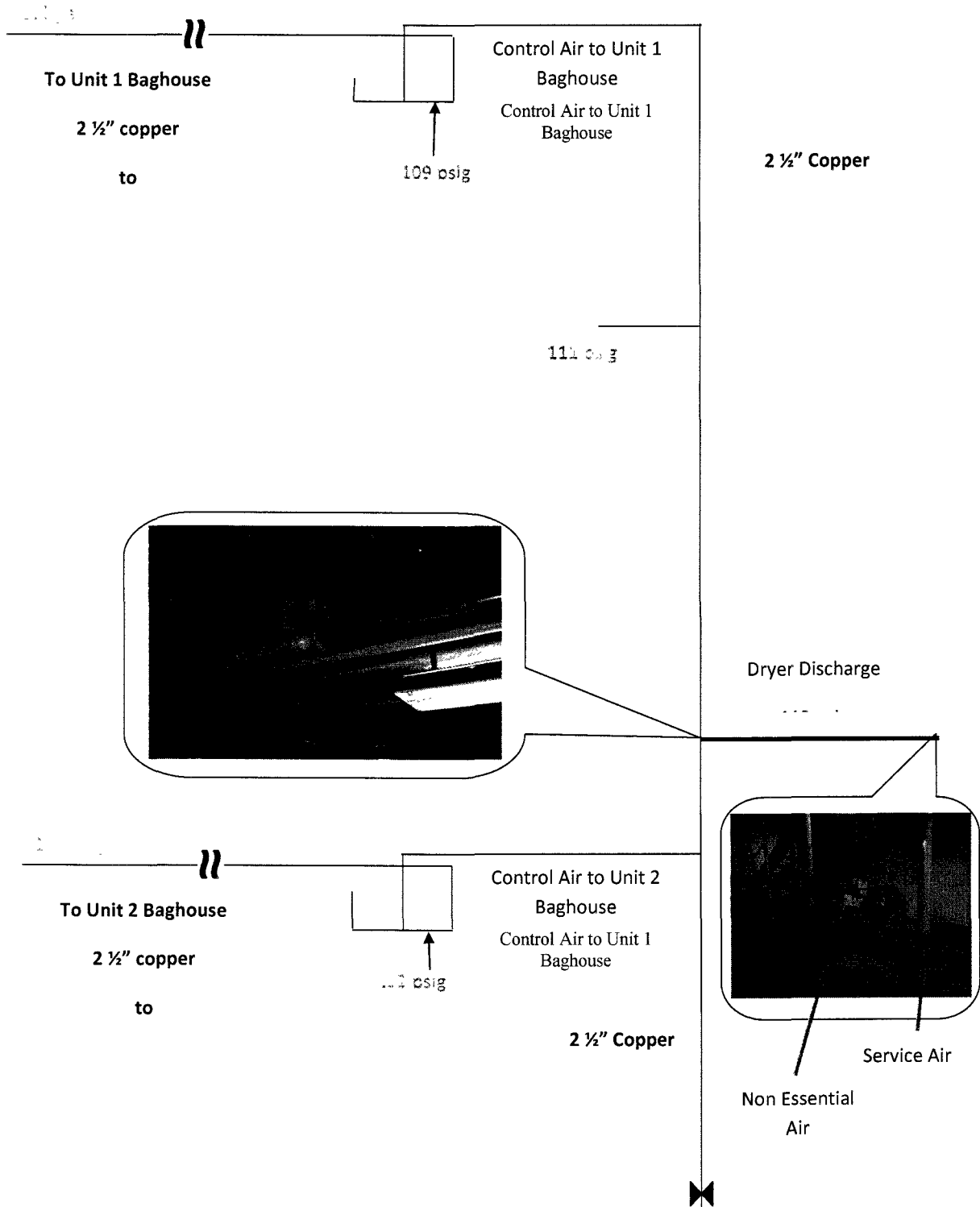






Figure 26. Transfer Building #2  
IPSC  
Transfer Building 2

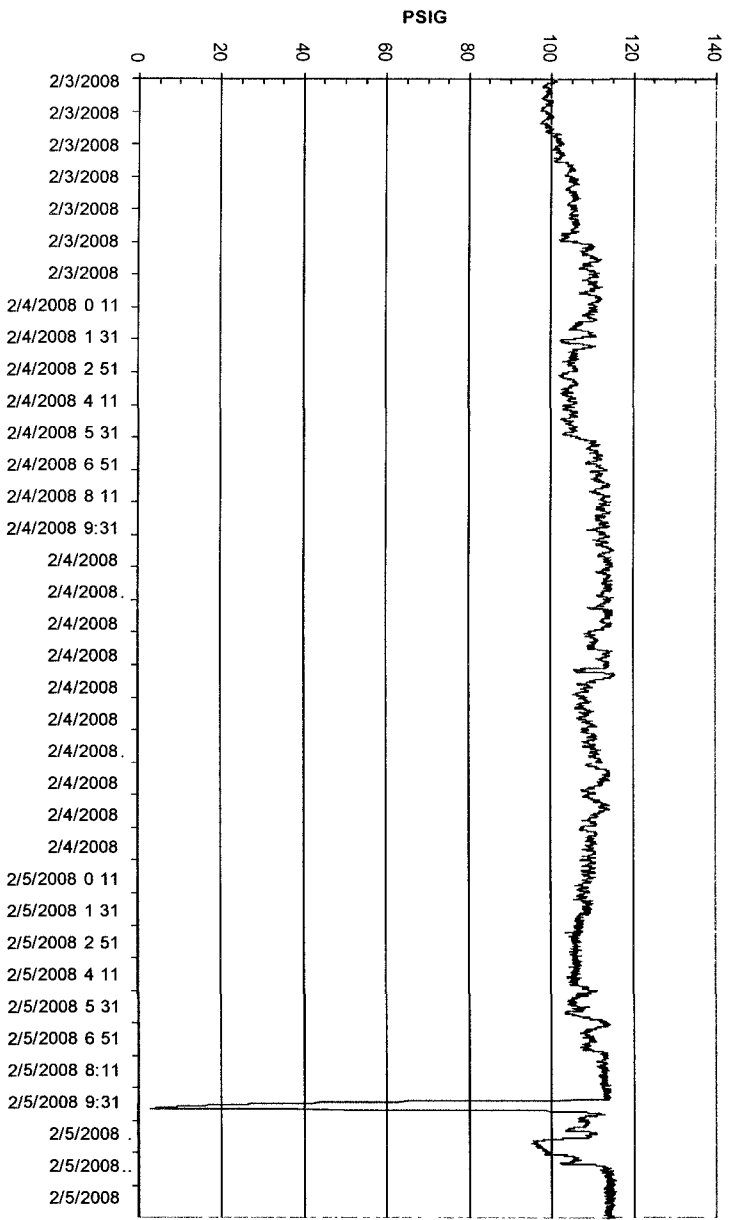


Figure 27. Transfer Building #1  
IPSC  
Transfer Building 1

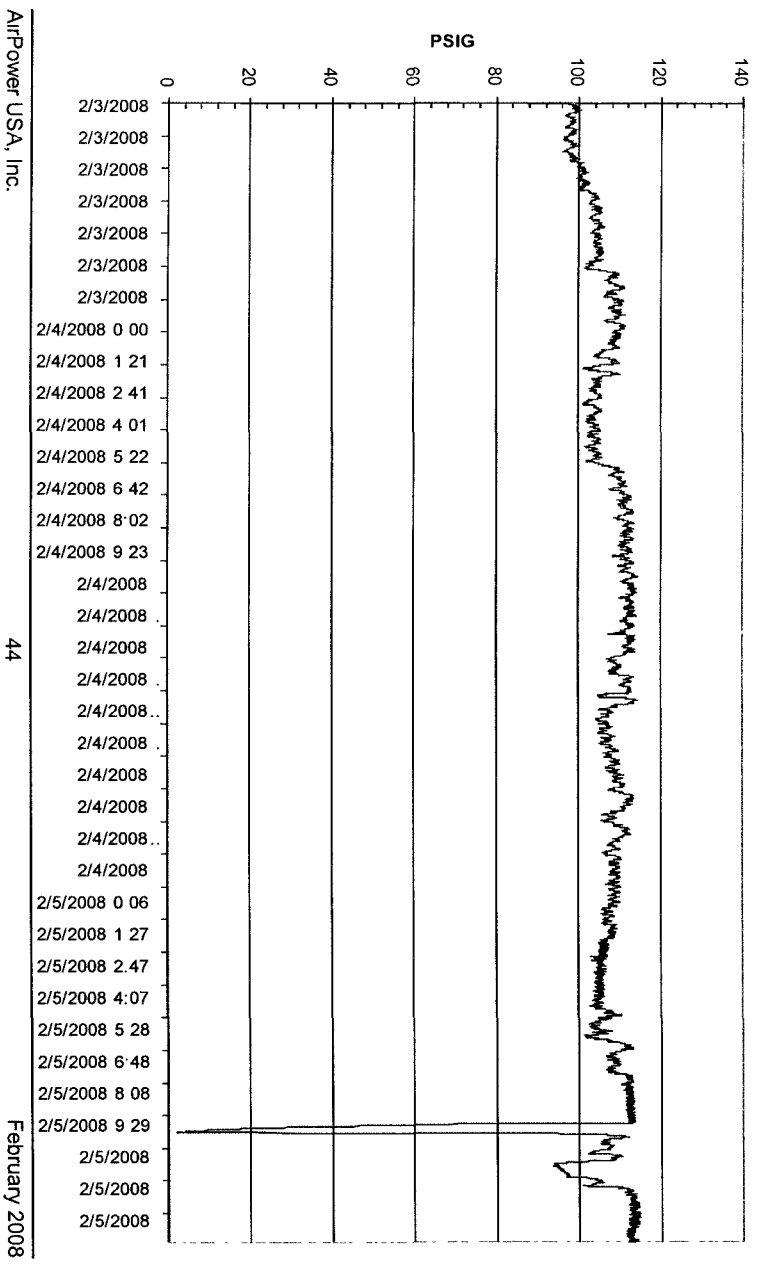






Figure 32. Water Treatment  
IPSC  
Water Treatment

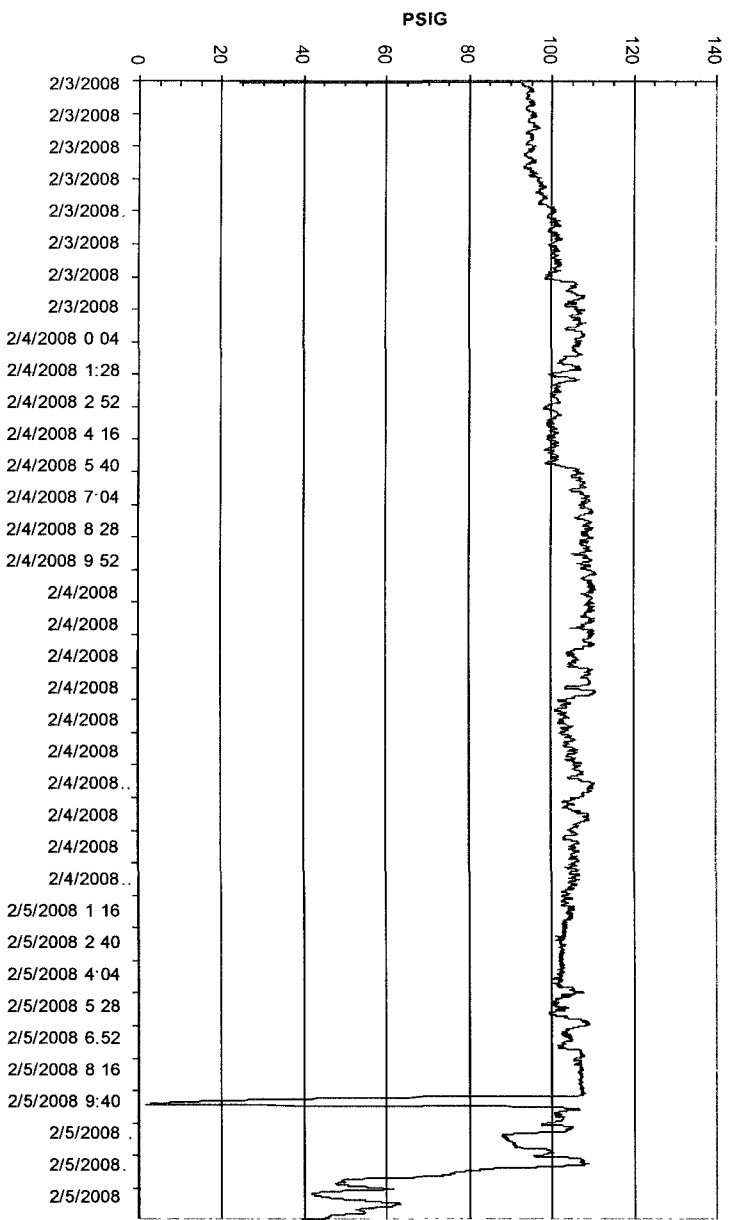


Figure 33. Boiler Sump Transfer  
IPSC  
BLR Sump Transfer

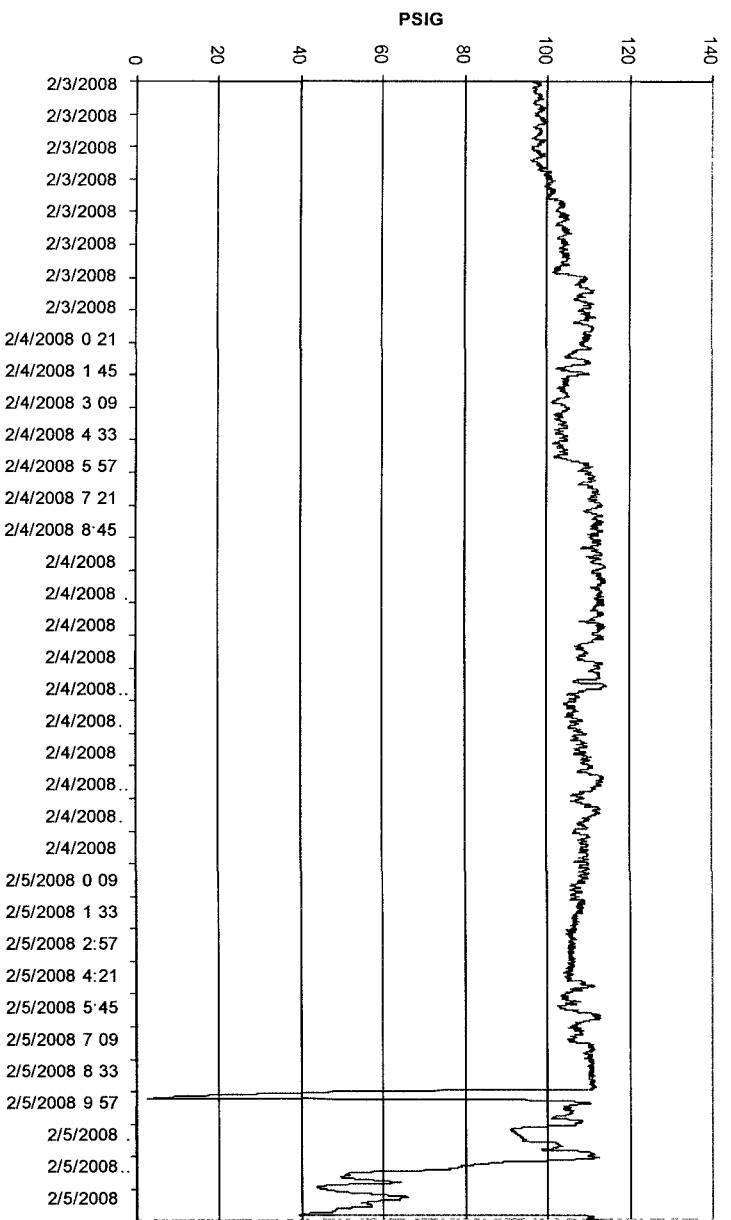


Figure 34. Igniter Boost Receiver  
IPSC  
Ignition Boost

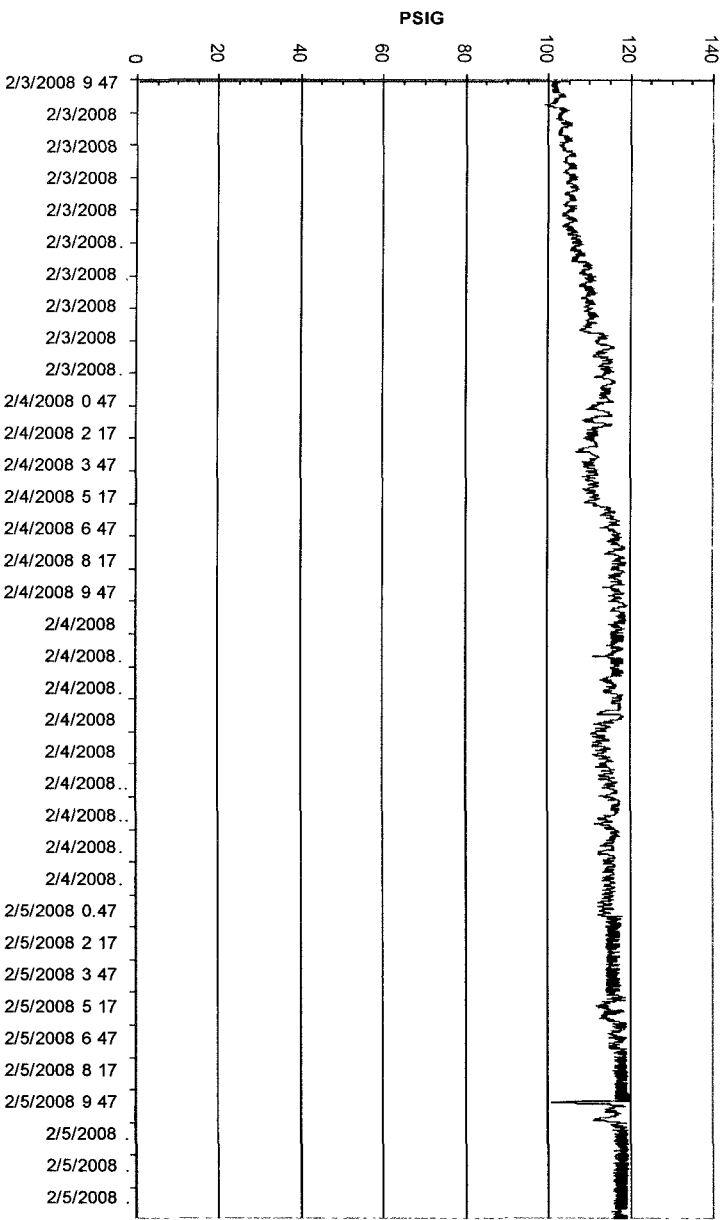
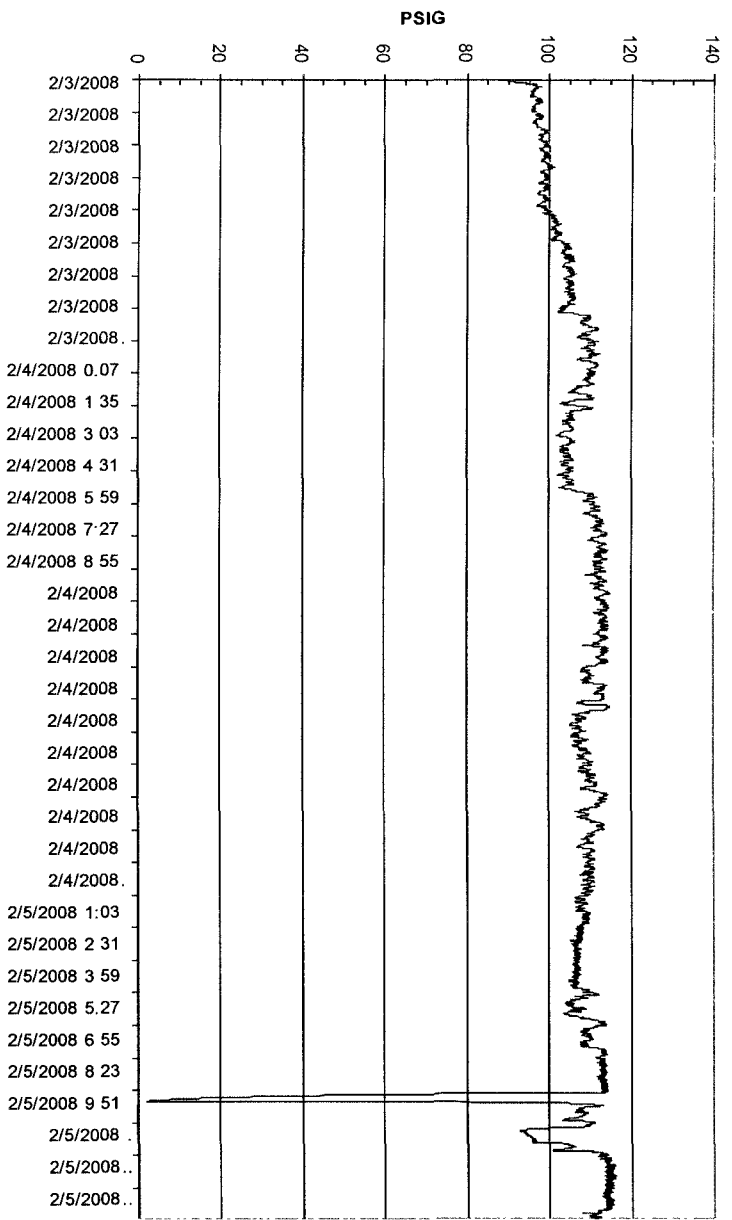


Figure 35. Lime Preparation Receiver  
IPSC  
Lime Prep





## 4.2 PROCESS REGULATORS

### Background

There are additional direct power cost savings to be obtained if the plant can continue to lower the overall system operating pressure. A steady delivered system pressure will allow follow-up programs at each process or point of use to establish the lowest overall effective pressure. This will enhance productivity, quality, and continue to reduce air usage and production costs.

The cornerstone of any effective demand-side air conservation program is to identify and operate at the lowest acceptable operating pressure required at all of the various production sectors and operating units in the plant. This should be a continuing program and part of any air system training or usage awareness program.

Some regulators are probably set at a higher-than-necessary feed pressure for an individual process, while others may be set wide open to full header pressure. Key questions to consider include: is there a minimum effective pressure established for each point of use for each production run? And, if so, is it being adhered to?

In this type of operation, it is very important that the actual inlet pressure for each process be known and that the lowest effective pressure be held steady for the proper product quality. Installation of storage bottles downstream of the regulator may be needed to “close up” the pressure readings at rest and at operation and offset regulator delay.

- ☒ **PHASE 2 RECOMMENDATION** – Review all regulated operations to establish lowest effective settings.

## 4.3 DUST COLLECTORS

### Background

Proper operation of dust collectors is critical to minimizing cost and maximizing system effectiveness. There are many sizes and most, if not all, use a pulse of compressed air controlled by a timer. The timers are generally set by the operators to what they believe is appropriate for proper cake removal and bag life.

In a dust collection system, the dust is collected on the bag or fingers and when the cake of dust is of appropriate thickness and structure – a pulse or pulses of compressed air is used to hit or shock the bag and knock the cake off.

When the cake is removed correctly from the dust collector, the system removes dust from its assigned environment and has a normal bag life. When the cake is not removed effectively, the dust collector does not remove dust effectively from its assigned environment and the bag life can be significantly shortened.

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Dust collection system designs specify the air inlet pressure to the manifold and pulse valves necessary for effective dust removal. The pulse valve sends a given volume or weight of air to the bag at a predetermined velocity to strike and clear the cake. The actual amount or weight of air is dependent upon the pulse nozzle being fed compressed air at a pre-determined and steady pressure.

The dust collector must receive the correct pressure (or close to it) and a steady repeatable pressure level for each pulse, particularly if timers are used to control the pulses. The operator may experiment to find the “right timing sequence” at a desired feed pressure. But if this pressure varies, then performance may not be satisfactory.

A problem that often occurs (short bag life) usually comes from the pulsers hitting the bag when the cake is not ready to flake off or the cake has gone too long between pulsing and grown too thick and heavy to clean effectively. This causes not only short bag life but very poor performance. There are usually several basic causes for this:

- Incorrect timer settings for the operating conditions. The actual requirement for the optimum timer setting may well change as various product runs change or even

seasonally. These settings have to be set carefully to begin with and monitored regularly.

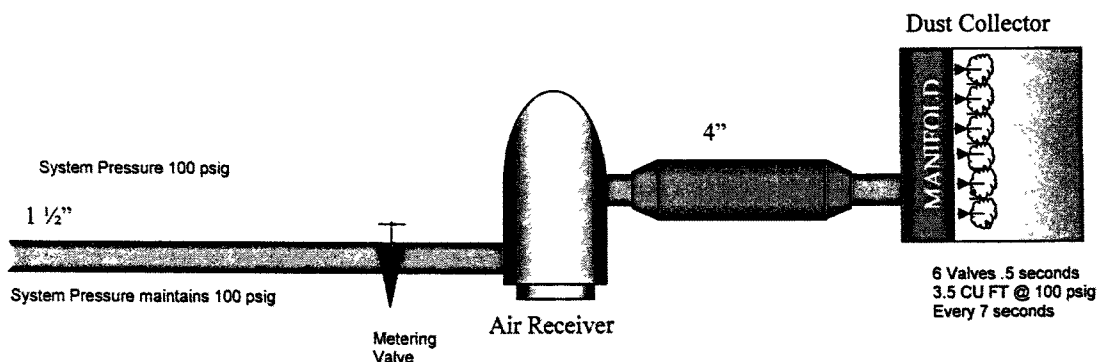
- Lack of sufficient storage or compressed air supply near the inlet manifold to supply the required pulse air without collapsing the inlet pressure. With too low an inlet pressure, the mass weight of the air pulse is too low, which then becomes ineffective in removing the cake.
- Too small a feed line to the dust collector will have the same effect as lack of air supply.
- Too small or incorrect regulator, which is unable to handle the required "Rate of Flow" required by the dust collectors.

All of these are installations or system situations that cause restricted air flow. They occur because, prior to the installation or prior to some operational change, the proper "rate of flow" was not identified for the dust collection action. Feed line sizing, regulator sizing, and air supply all require an identified "rate of flow." You cannot use "average flow rate."

"Flow rate" is the average flow or compressed air in cubic feet per minute either required by a process or delivered to the system. "Rate of flow" is the actual rate of flow of compressed air demand in cubic feet per minute. Even relatively small air demands in cubic feet can have a very high "rate of flow", if they occur over a very short time period. Dust collectors have this characteristic.

The sequence controllers can have a very significant impact on the required "rate of flow." For example, pictured here is a dust collector system, which has six pulsing valves that use 3.5 cu ft over 1/2 second for each pulse.

***Problem: "Flow" / "Rate of Flow"***



The impact of these two different "rates of flow" would show similar differences in regulator sizing, etc., if they are used on the feed line flow. The high flow velocities entering the manifold and controls for the pulse valves will create extra pressure loss through the balance affecting the performance of the pulse cleaner. The same sort of effect would show up in air receiver sizing to minimize system and feed line pressure drop if that is a question.

### Typical Sizing (Each Valve Uses 3.5 scfm/pulse – 6 Valves on Collector)

| Rate of Flow & Sizing with One Valve Hitting Every 7 Seconds  | Rate of Flow & Sizing with Six Valves Hitting Every 7 Seconds  |
|---|--|
| Rate of Flow =<br>(1) x (3.5) = 3.5 x 60 ÷ .5 = 420 scfm  | Rate of Flow =<br>(6) x (3.5) 21 x 60 ÷ .5 = 2,520 scfm  |
| The line size recommendation from the air supply to the dust collector = 90 psig line pressure = 2" to 3"   | The line size recommendation from the air supply to the dust collector – 90 psig line pressure = 4" to 6"  |
| <ul style="list-style-type: none"> <li>A 2" feed line will handle the 420 cfm flow at 90 psig line pressure with a velocity of 43 fps, which is about as high as it should go</li> <li>A 3" feed line will handle the 420 cfm flow at 90 psig with a velocity of about 19 fps – very conservative</li> <li>A 2" line would have a pressure loss of about 1 psid every 100' @ 420 scfm flow, which may be acceptable depending on feed line design and length</li> <li>A 3" line would have pressure loss of less than .10 psid per 100' @ 420 scfm flow, which should be very acceptable</li> </ul> | <ul style="list-style-type: none"> <li>A 5" feed line will handle the 2,520 cfm flow at 90 psig line pressure with a velocity of about 43 fps</li> <li>A 6" feed line will handle the 2,520 cfm flow at 90 psig line pressure with a velocity of about 30 fps, which is conservative in this application</li> <li>A 2" line at 2,520 cfm would have a minimum pressure loss of 30-50 psid, depending on timing and turbulence. This would be completely unacceptable</li> <li>A 4" line would have a pressure loss of about 1.1 to 1.2 psid per 100' @ 90 psig and combined with moderate velocity should be acceptable depending on the length and design of the feed line</li> <li>A 6" line would have a minimum pressure loss of .15 to .20 @ 90 psig with very low velocities and should be acceptable with "normal" installations</li> </ul> |

We recommend that every feed line has a quality pressure gauge installed near the dust collector entry. Observe the pressure gauge, which the pulser hits – if the pressure drop is too high (over 10-20 psig), start looking for the cause. Get the specification on the dust collector, cfm per pulse, feed line pressure time per pulse, cycle time between pulses, etc. Calculate the rate of flow, check line size and storage. If additional storage is required, this can be calculated by the following formula:

For example, 2,520 rate of flow @ .5 seconds flow with 4 psig allowable pressure loss.

**Size Air Receiver:**  $T_{min} = (V) (P_2 - P_1)$   
(CFM) (14.5)

Net Rate of Flow 2,520 cfm

$P_1$  – Rest Pressure 100 psig

$P_2$  – Allowable Drop 96 psig (4 psig)

$$T_{sec} = \frac{(V) (4) (60)}{.5 \text{ sec } (2,520)(14.4)} = .5 \text{ sec} = \frac{240 V}{36,540}$$

$$240 V = 18,270 V = 76 \text{ cu ft} \times 7.48 = 570 \text{ gal or more}$$

#### Refill Rate of Flow:

Time allowed – 6 seconds

$$21 \text{ cu ft} \times 60 \text{ seconds} \div 6 \text{ seconds} =$$

210 cfm rate of flow

**Effect on Header:** Negligible

We have used storage to convert a high rate of flow to a low rate of flow and eliminate system pressure collapse

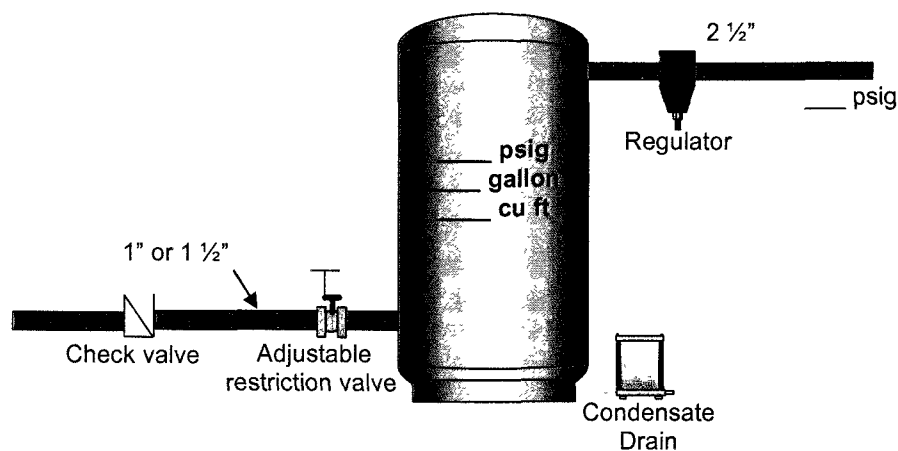
Significant amounts of air (10 to 15 cfm or more) can be lost when the control diaphragm and/or connections fail. Such leaks are very difficult to find and repair.

Proper sizing and installation of appropriate storage for dust collectors offers opportunities to convert high volume short term demand to lower average rate of flow.

Install an appropriate sized receiver near the process (dust collector, etc.). System air at 90 psig\* from inlet and regulate to each feed line to dust collector. Size the regulator to handle appropriate scfm rate of flow with minimum inlet pressure of 92 psig. Install check valve and adjustable restriction valve on inlet line to air receiver.

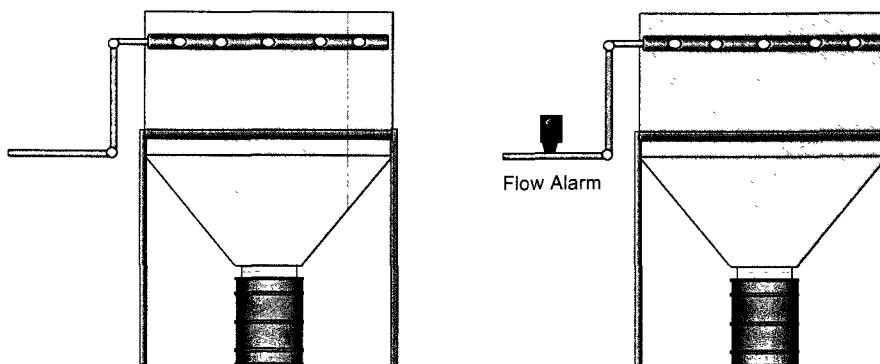
(\* Pressure must exceed the minimum for process.)

For example, Pulse is 3.5 cfm ½ sec every 7 seconds (see preceding page)



Adding appropriate storage may not only be a direct energy issue but one of air quality. Proper control of the dust collectors will protect surrounding systems from falling pressure at nozzle blow. This should also enhance the dust collector performance and extend bag life.

Dust collectors are a significant source of leaks that are hard to detect. Often the pulse control diaphragms leak. An electronic air flow alarm can signal this problem visually and remotely.



## Current Application

Some of the dust collector feeds appear to be marginally sized, and each does not have an air receiver between it and the collector. Observation of the operation and discussions with plant personnel indicate the demand controls are working well and the bags sloughing off properly. There does not appear to be a problem of pulling low pressure in surrounding lines but there does appear to be some problem pulling the inlet pressure down to the collector drain at the pulse (see the following data collected at the Limestone unload).

The procedures shown on the preceding pages indicate how to size a proper volume receiver to avoid this.

The Limestone unloading dust collectors may operate 4-6 hours a day and the conveyors 14-15 hours per day. However, our observation indicates:

Due to the successful DP control system, the actual cleaning time is much less. The pulses appear to be well controlled.

## Summary

There does not appear to be a significant air saving opportunity here at this time. However, there does appear to be some action that can be taken to stabilize the actual feed pressure to the pulses and avoid today's turndown.

## Observed Operation of Limestone Unloading Dust Collector

Date: 6 February 2008  
Time: 10:30 – 11:30 am  
Inlet: Press to dust collector  
1" feed line; 35' long – regulator and filter – set to 90 psig

The plant was advised to run at 80 psig pulse pressure to avoid bag premature deterioration with the Teflon-coated bag. The regulator was set higher to avoid low pressure shut off.

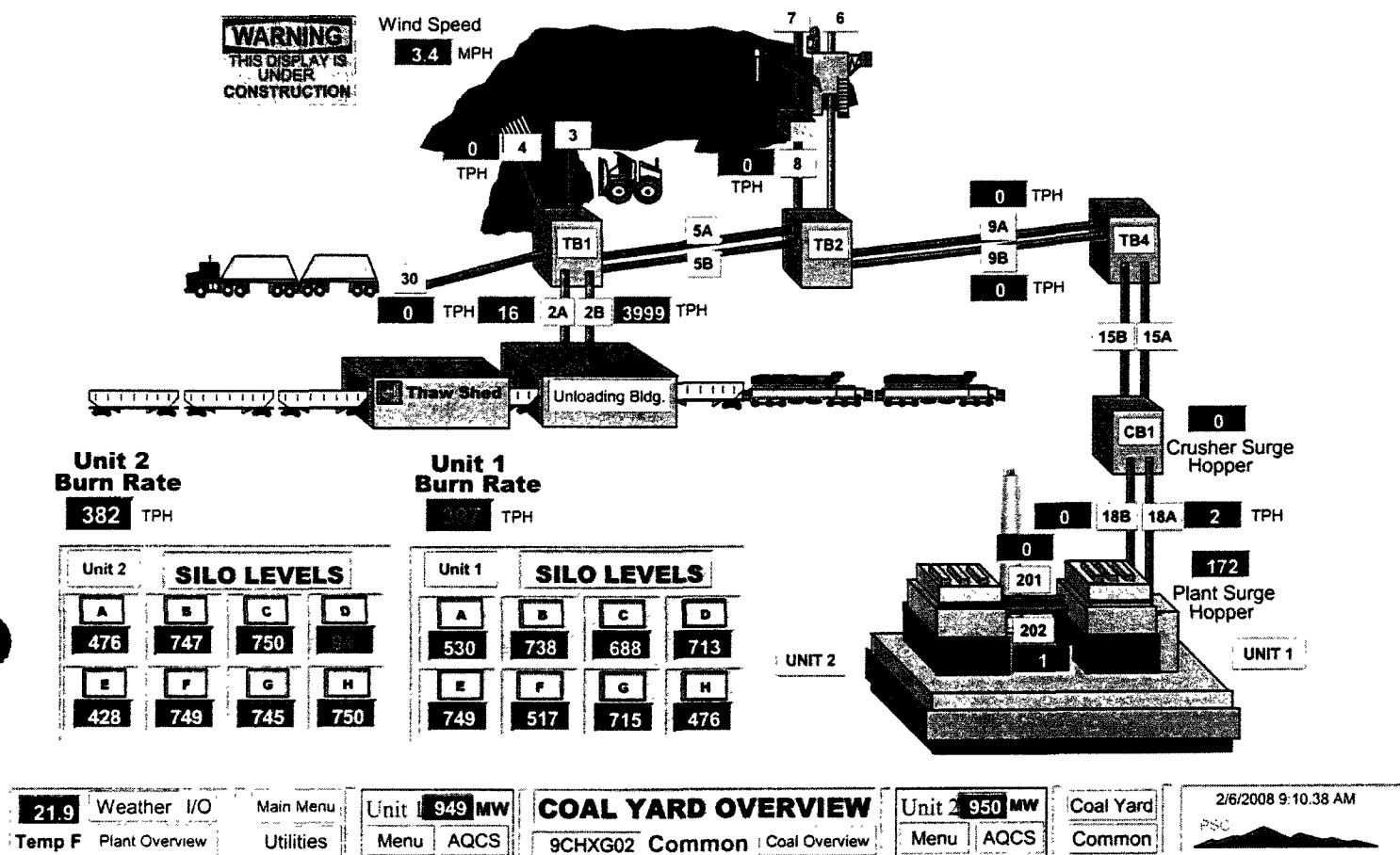
Inlet pressure fell at pulse (6 pulsers):

|    | <u>Drop</u> |    | <u>Actual Inlet</u>            |
|----|-------------|----|--------------------------------|
| #1 | 30 psig     | to | 60 psig                        |
| #2 | 30 psig     | to | 60 psig                        |
| #3 | 38 psi      | to | 52 psig (possible "blown bag") |
| #4 | 39 psig     | to | 51 psig (possible "blown bag") |
| #5 | 30 psig     | to | 60 psig                        |
| #6 | 30 psig     | to | 60 psig                        |

The above data shows that actual inlet pressure to the pulser is not the recommended 80 psig, but 60 psig. This may be having a negative impact on bag filter integrity. We recommend investigation when convenient to see if this is or is not a problem to be addressed. If it is a

problem, the methodology for correction is in the preceding sample or we will be happy to work with you on this.

Figure 37. Coal Yard Overview



#### 4.4 LEAK IDENTIFICATION AND REPAIR

Most plants can benefit from an ongoing leak management program. Generally speaking, the most effective programs are those that involve the production supervisors and operators working in concert with the maintenance personnel. Accordingly, it is suggested that all programs consist of the following:

- **Short Term** – Set up a continuing leak inspection by Maintenance Personnel so that for a while, each primary sector of the plant is inspected once each quarter to identify and repair leaks. A record should be kept of all findings, corrective measures, and overall results. The PROJECT COST SECTION below includes current price quotes for ultrasonic leak locator equipment.
- **Long Term** – Consider setting up programs to motivate the operators and supervisors to identify and repair leaks. One method that has worked well with many operations is to monitor the air flow to each department and make each department responsible for identifying its air usage as a measurable part of the operating expense for that area. This usually works best when combined with an effective in-house training, awareness, and incentive program.

With a plant of this type, an effective leak management program could save 1,500 cfm or the equivalent of repairing 500 leaks averaging 3 cfm each. On a percentage basis, this leak level is about the same as leak levels in other plants. Repairing leaks totaling 1,500 cfm translate into an annual savings of \$152,505 per year.

##### Compressed Air Leak Survey

A survey of compressed air leaks was conducted at the plant and 174 leaks were identified, quantified, tagged, and logged. Potential savings totaled 855 cfm for the 174 leaks that were identified.

We recommend an ultrasonic leak locator be used to identify and quantify the compressed air leaks. We use either a VXP AccuTrak manufactured by Superior Signal or a UE Systems Ultraprobe 2000.

Shutting off the air supply to these leaks when the area is idle would save significant energy use. Reducing the overall system pressure would also reduce the impact of the leaks, when air to the machine cannot be shut off. Repairing the leaks can save additional energy. The savings estimates associated with a leak management program are based on the unloading controls of the compressors being able to effectively translate less air flow into lower cost.

With a few minor exceptions, most of the leaks could not have been found without the use of an ultrasonic leak detector and a trained operator. Leak locating during production time with the proper equipment is very effective and often shows leaks that are not there when idle. However, a regular program of inspecting the systems in “off hours” with “air powered up” is also a good idea. In a system such as this one, some 90 to 95% of the total leaks will be in the use of the machinery, not in the distribution system.

The area surveyed in the leak study included a great deal of high background ultrasound noise that shields many of the smaller leaks. In continuing the leak management program, plant staff

should perform leak detection during non-production hours in order to eliminate some of the high ultrasonic background noise.

- ☒ **RECOMMENDED PROJECT (#7)** – Implement ongoing leak identification and repair program with ultrasonic locators. Repair all tagged leaks listed on the following page.

|   |                 |
|---|-----------------|
| Estimated reduction of air flow with proposed project     | 855 cfm         |
| Recoverable savings from air flow reduction [Section 2.3] | \$84.53 /cfm yr |
| Annual electric cost savings with proposed project        | \$72,273 /year  |
| Cost of leak detection equipment (if required)            | \$2,800         |
| Number of leaks   | 178             |
| Estimated cost of leak repairs (\$100 per leak)           | \$17,800        |
| Total project cost (materials and installation)           | \$20,600        |

- ☒ **PHASE 2 RECOMMENDATION** – Continue an aggressive leak tagging and repair program. Quantify and value the leak(s) and report to management on a predetermined regular basis.

#### Leak List

| <b>TAG</b> | <b>LOCATION</b> | <b>DESCRIPTION</b>             | <b>EST. SIZE</b> | <b>EST. CFM.</b> |
|------------|-----------------|--------------------------------|------------------|------------------|
| 3829       | COMPRESSOR ROOM | COMPRESSOR 1A A/C BLOWDN & VLV | LARGE            | 20               |
| 3829       | COMPRESSOR ROOM | 1D COMPRESSOR A/C DRAIN        | VERY LG          | 70               |
| 3830       | UNIT 2 BAGHOUSE | A CASING CYL 1087              | LARGE            | 15               |
| 3831       | UNIT 2 BAGHOUSE | A CASING CYL 1103              | SMALL            | 3                |
| 3832       | UNIT 2 BAGHOUSE | A CASING CYL 1104              | SMALL            | 3                |
| 3833       | UNIT 2 BAGHOUSE | A CASING PURGE AIR             | SMALL            | 3                |
| 3834       | UNIT 2 BAGHOUSE | CYL LEAK--C16 CYLINDER 3147    | SMALL            | 4                |
| 3835       | UNIT 2 BAGHOUSE | CYL LEAK--C13 CYLINDER 3120    | SMALL            | 2                |
| 3836       | UNIT 2 BAGHOUSE | CYL LEAK--C12 CYLINDER 3110    | SMALL            | 3                |
| 3837       | UNIT 2 BAGHOUSE | CYL LEAK--C11 CYLINDER 3101    | SMALL            | 3                |
| 3838       | UNIT 2 BAGHOUSE | CYL LEAK--C3 CYLINDER 3030     | SMALL            | 2                |
| 3839       | UNIT 2 BAGHOUSE | CYL LEAK--C1 CYLINDER 3012     | SMALL            | 2                |
| 3840       | UNIT 2 BAGHOUSE | CYL LEAK--C9 CYLINDER 3083     | SMALL            | 4                |
| 3841       | UNIT 2 BAGHOUSE | CYL LEAK--B1 CYLINDER 2012     | SMALL            | 2                |
| 3842       | UNIT 2 BAGHOUSE | CYL LEAK--B1 CYLINDER 2011     | SMALL            | 2                |
| 3843       | UNIT 2 BAGHOUSE | CYL LEAK--B11 CYLINDER 2101    | SMALL            | 3                |
| 3844       | UNIT 2 BAGHOUSE | CYL LEAK--B11 CYLINDER 2102    | SMALL            | 2                |
| 3845       | UNIT 2 BAGHOUSE | CYL LEAK--B3 CYLINDER 2030     | SMALL            | 3                |
| 3846       | UNIT 2 BAGHOUSE | CYL LEAK--B12 CYLINDER 2110    | SMALL            | 2                |

|      |                 |                             |       |   |
|------|-----------------|-----------------------------|-------|---|
| 3847 | UNIT 2 BAGHOUSE | CYL LEAK--B12 CYLINDER 2111 | SMALL | 3 |
| 3848 | UNIT 2 BAGHOUSE | CYL LEAK--B4 CYLINDER 2038  | SMALL | 3 |
| 3849 | UNIT 2 BAGHOUSE | CYL LEAK--B13 CYLINDER 2119 | SMALL | 4 |
| 3850 | UNIT 2 BAGHOUSE | CYL LEAK--B13 CYLINDER 2120 | SMALL | 3 |
| 3851 | UNIT 2 BAGHOUSE | CYL LEAK--B14 CYLINDER 2128 | SMALL | 2 |
| 3852 | UNIT 2 BAGHOUSE | CYL LEAK--B14 CYLINDER 2129 | SMALL | 2 |
| 3853 | UNIT 2 BAGHOUSE | CYL LEAK--B7 CYLINDER 2065  | SMALL | 2 |
| 3854 | UNIT 2 BAGHOUSE | CYL LEAK--B15 CYLINDER 2133 | SMALL | 4 |
| 3855 | UNIT 2 BAGHOUSE | CYL LEAK--B8 CYLINDER 2075  | SMALL | 2 |
| 3856 | UNIT 2 BAGHOUSE | CYL LEAK--A8 CYLINDER 1075  | SMALL | 2 |
| 3857 | UNIT 2 BAGHOUSE | CYL LEAK--A15 CYLINDER 1138 | MED   | 5 |
| 3858 | UNIT 2 BAGHOUSE | CYL LEAK--A14 CYLINDER 1129 | SMALL | 3 |
| 3859 | UNIT 2 BAGHOUSE | CYL LEAK--A13 CYLINDER 1120 | SMALL | 3 |
| 3860 | UNIT 2 BAGHOUSE | CYL LEAK--A13 CYLINDER 1119 | SMALL | 2 |
| 3861 | UNIT 2 BAGHOUSE | CYL LEAK--A4 CYLINDER 1038  | SMALL | 2 |
| 3862 | UNIT 2 BAGHOUSE | CYL LEAK--A10 CYLINDER 1093 | SMALL | 3 |
| 3863 | UNIT 2 BAGHOUSE | CYL LEAK--A2 CYLINDER 1020  | SMALL | 3 |
| 3864 | UNIT 2 BAGHOUSE | 1A01 CYL 1011               | SMALL | 3 |
| 3865 | UNIT 2 BAGHOUSE | C-CASING CYL 3085--SOL EXH  | SMALL | 2 |
| 3866 | UNIT 2 BAGHOUSE | C-CASING CYL 3094--TOP      | MED   | 5 |
| 3867 | UNIT 2 BAGHOUSE | C-CASING CYL 3114           | SMALL | 3 |
| 3868 | UNIT 2 BAGHOUSE | C-CASING CYL 3151           | SMALL | 3 |
| 3869 | UNIT 2 BAGHOUSE | C-CASING CYL 3076           | SMALL | 3 |
| 3870 | UNIT 2 BAGHOUSE | C-CASING CYL 3070           | SMALL | 3 |
| 3871 | UNIT 2 BAGHOUSE | C-CASING CYL 1007 PURGE     | SMALL | 3 |
| 3872 | UNIT 2 BAGHOUSE | C-CASING CYL 3033           | MED   | 6 |
| 3873 | UNIT 2 BAGHOUSE | C-CASING CYL 3022           | SMALL | 2 |
| 3874 | UNIT 2 BAGHOUSE | C-CASING REG--AIR HORN AIR  | SMALL | 2 |
| 3875 | UNIT 2 BAGHOUSE | C CASING CYL 1C01 TOP       | SMALL | 3 |
| 3876 | UNIT 2 BAGHOUSE | B CASING CYL 2150           | SMALL | 2 |
| 3877 | UNIT 2 BAGHOUSE | B CASING CYL 2140           | SMALL | 2 |
| 3878 | UNIT 2 BAGHOUSE | B CASING 3/4" FILTER        | MED   | 5 |
| 3879 | UNIT 2 BAGHOUSE | B CASING CYL 2122           | SMALL | 3 |
| 3880 | UNIT 2 BAGHOUSE | B CASING CYL 2121           | SMALL | 3 |
| 3881 | UNIT 2 BAGHOUSE | B CASING CYL 2115           | SMALL | 3 |
| 3882 | UNIT 2 BAGHOUSE | B CASING CYL 2113           | SMALL | 3 |
| 3883 | UNIT 2 BAGHOUSE | B CASING CYL 2112           | SMALL | 2 |
| 3884 | UNIT 2 BAGHOUSE | B CASING CYL 2104           | SMALL | 2 |
| 3885 | UNIT 2 BAGHOUSE | B CASING CYL 2096           | SMALL | 2 |
| 3886 | UNIT 2 BAGHOUSE | B CASING CYL REV AIR        | MED   | 5 |
| 3887 | UNIT 2 BAGHOUSE | B CASING CYL 2014           | SMALL | 3 |
| 3888 | UNIT 2 BAGHOUSE | B CASING CYL 2040           | MED   | 6 |
| 3889 | UNIT 2 BAGHOUSE | B CASING CYL 2041           | SMALL | 3 |
| 3890 | UNIT 2 BAGHOUSE | B CASING CYL 2059A          | SMALL | 3 |
| 3891 | UNIT 2 BAGHOUSE | B CASING CYL 2068           | SMALL | 3 |
| 3892 | UNIT 2 BAGHOUSE | A CASING CYL 1014           | SMALL | 3 |
| 3893 | UNIT 2 BAGHOUSE | A CASING CYL 1032--SOL VLV  | MED   | 5 |
| 3894 | UNIT 2 BAGHOUSE | A CASING CYL 1050           | SMALL | 3 |

|      |                                |   |        |    |
|------|--------------------------------|---|--------|----|
| 3895 | UNIT 2 BAGHOUSE                | A CASING CYL 1068                       | LARGE  | 10 |
| 3896 | UNIT 2 BAGHOUSE                | A CASING CYL 1076                       | SMALL  | 2  |
| 3897 | UNIT 2 BAGHOUSE                | A CASING CYL 1130                       | MED    | 5  |
| 3898 | UNIT 2 BAGHOUSE                | A CASING CYL 1115                       | MED    | 5  |
| 3899 | UNIT 2 BAGHOUSE                | A CASING CYL 1114                       | SMALL  | 3  |
| 3899 | UNIT 2 BAGHOUSE                | A CASING CYL 1105                       | SMALL  | 3  |
|      | <b>Unit 1 Casing Bag House</b> |   |        |    |
| 3900 | 1A stdnd 35                    | ZSL block vent/lower                    | small  | 3  |
| 3901 | 1A bypas #3                    | mositure trap/site glass                | small  | 3  |
| 3902 | 1A bypass north wall           | mositure trap/bottom fitting            | small  | 3  |
| 3903 | Reverse Air Relief damper      | cyld packing/when up position           | small  | 3  |
| 3904 | stand 1 cyld                   | packing seal/when up position           | small  | 4  |
| 3905 | stand 2 cyld                   | packing seal/when up position           | small  | 4  |
| 3906 | stand 3                        | muffler filter/upper muffler            | medium | 5  |
| 3907 | stand 9 cyld                   | packing seal/when up position           | small  | 3  |
| 3908 | stand 21                       | muffler filter/lower muffler            | small  | 3  |
| 3909 | stand 22                       | muffler filter/upper muffler            | small  | 3  |
| 3910 | stand 23 cyld                  | packing seal/when up position           | small  | 4  |
| 3911 | stand 32                       | ZSL block vent/upper vent               | medium | 5  |
| 3912 | stand 34 cyld                  | packing seal/when up position           | small  | 3  |
| 3913 | stand 35 cyld                  | packing seal/when up position           | medium | 5  |
| 3914 | stand 42                       | muffler filter/upper muffler            | medium | 5  |
|      | stand 42 cyld                  | packing seal/when up position           | large  | 10 |
| 3915 | stand 51 cyld                  | packing seal/when up position           | small  | 3  |
| 3916 | stand 55 cyld                  | packing seal/when up position           | medium | 4  |
| 3917 | stand 57                       | ZSL block vent/upper vent               | small  | 3  |
| 3918 | stand 58                       | muffler filter/upper muffler            | small  | 3  |
| 3919 | stand 59                       | muffler filter/upper muffler            | small  | 3  |
| 3920 | FF compt 1A6                   | ZSH block vent/left cyld                | small  | 4  |
| 3921 | FF compt 1B14                  | ZSH block vent/left cyld                | small  | 3  |
| 3922 | FF compt 1B5                   | ZSH block vent/left cyld                | medium | 5  |
| 3923 | FF compt 1B4                   | ZSH block vent/left cyld                | medium | 6  |
| 3924 | FF compt 1B12                  | ZSH block vent/right cyld               | small  | 3  |
| 3925 | FF compt 1B11                  | ZSH block vent/right cyld               | medium | 4  |
| 3926 | FF compt 1B10                  | ZSH block vent/right cyld               | small  | 3  |
| 3927 | FF compt 1C5                   | ZSH block vent/right cyld               | small  | 3  |
| 3928 | B Casing relief cyld           | packing seal/when up position           | medium | 4  |
| 3929 | stand 1 cyld                   | ZSL block vent/lower block vent         | small  | 3  |
| 3930 | stand 2                        | ZSL block vent/upper vent               | small  | 3  |
| 3931 | stand 6                        | ZSL block vent/upper vent               | small  | 3  |
| 3932 | stand 9                        | muffler filter/lower muffler            | small  | 4  |
| 3933 | STAND 9                        | ZSL block vent/lower vent               | small  | 3  |
| 3934 | stand 10                       | muffler filter/upper muffler            | medium | 5  |
| 3935 | stand 11 cyld                  | packing seal/when up position           | small  | 3  |
| 3936 | Between stand 12 & 13          | Lubricator/top piece                    | medium | 8  |
| 3937 | stand 14 cyld                  | packing seal/when up position           | medium | 6  |
| 3938 | stand 19                       | electric solenoid/attaches to air block | small  | 2  |

|        |                         |  |        |    |
|--------|-------------------------|--|--------|----|
| 3939   | stand 20                | muffler filter/lower muffler                           | small  | 3  |
| 3940   | purge Hdr between 36/37 | moisture trap/bottom of bowl                           | medium | 8  |
| 3941   | stand 34                | ZSL block vent/lower block vent                        | small  | 2  |
| 3942   | stand 38                | ZSL block vent/lower block vent                        | medium | 4  |
| 3943   | stand 41                | ZSL block vent/upper block vent                        | small  | 3  |
| 3944   | stand 45                | muffler filter/lower muffler                           | small  | 3  |
| 3945   | stand 49                | muffler filter/lower muffler                           | small  | 3  |
| 3946   | stand 50                | muffler filter/upper muffler                           | small  | 2  |
| 3947   | stand 50                | ZSL block vent/lower block vent                        | small  | 3  |
| 3948   | stand 54                | muffler filter/upper muffler                           | small  | 3  |
| 3950   | stand 54                | ZSL block vent/upper block vent                        | small  | 2  |
| 3951   | stand 60                | ZSL block vent/lower block vent                        | small  | 2  |
| 3952   | stand 62                | muffler filter/upper muffler                           | medium | 6  |
| 3953   | stand 64                | ZSL block vent/upper block vent                        | small  | 2  |
| 3954   | Casing C Relief damper  | cyld packing/when up position                          | medium | 5  |
| 3955   | stand 8                 | ZSL block vent/upper block vent                        | small  | 2  |
| 3956   | Stand 12                | ZSL block vent/upper block vent                        | small  | 2  |
| 3957   | stand 16                | ZSL block vent/lower block vent                        | small  | 2  |
| 3958   | stand 31 cyld           | packing seal/when up position                          | small  | 3  |
| 3959   | stand 31                | muffler filter/upper muffler                           | small  | 3  |
| 3960   | stand 38                | muffler filter/upper muffler                           | large  | 15 |
| 3961   | stand 44                | ZSL block vent/upper block vent                        | small  | 2  |
| 3962   | stand 48                | muffler filter/lower muffler                           | small  | 3  |
| 3963   | stand 55 cyld           | muffler filter/upper muffler                           | medium | 5  |
| 3964   | bag house FA feeder 1C9 | solenoid valve body/                                   | small  | 2  |
| 3965   | FA feeder 1C10          | solenoid valve body/                                   | small  | 3  |
| 3966   | FA feeder 1C13          | solenoid valve body/                                   | small  | 4  |
| 3967   | FA feeder 1C16          | air cyld/over head                                     | small  | 2  |
| 3968   | Fa feeder 1C5           | solenoid valve body/                                   | small  | 3  |
| 3969   | FA feeder 1B15          | solenoid valve body/over head with rag on it           | large  | 15 |
| 3970   | FA feeder 1A12          | air cyld/overhead                                      | small  | 2  |
| 3971   | FA feeder 1A14          | air cyld/overhead                                      | small  | 3  |
| 3972   | FA feeder 1A15          | air cyld/overhead                                      | small  | 3  |
| 3973   | conveyor 18A-B          | Air gear box /(2) ball valve is venting                | large  | 15 |
| 3974   | Conveyor 18 A-B         | air brake/both ball valves venting 100% of time        | Large  | 15 |
| 3975   | Conveyor 9              | #1 air drop leg/Bad Ball valve leaking when on         | Medium | 5  |
| 3976   | Conveyor 5              | #2 air drop leg/Bad Ball valve leaking when on         | Medium | 5  |
| 3977   | Conveyor 5              | #3 air drop leg/Bad Ball valve leaking when on         | Medium | 4  |
| 3978   | Transfer Building 1     | Dust collector/Air valve accuator on top of silo       | Large  | 15 |
| 3979   | B train water treatment | 9WTD-ABV-227/middle of valve body                      | Small  | 3  |
| 3980   | A train water treatment | 9WTD-ANX-3A/regulator                                  | Small  | 3  |
| 3981   | A train water treatment | 9WTD-ABV-96/middle of valve body                       | Small  | 3  |
| 3982   | A train water treatment | 9WTD-ABV-25/Diaphragm                                  | Small  | 3  |
|        | Paint Shop              | moisture trap/cracked bowl                             | Small  | 4  |
| No Tag | Dust collector B        | Damper valve on silo/Air valve accuator on top of silo | Large  | 25 |
| No Tag | Dust collector D        | Damper valve on silo/Air valve accuator on top of silo | Large  | 25 |

|     |                     |   |        |     |
|-----|---------------------|---|--------|-----|
| A1  | Sludge C            | Thickner Tunnel/under stairs soleniod blowing     | Large  | 15  |
| A2  | Sludge Tank Bldg    | ABV 221/up stairs hose                            | Medium | 4   |
| A3  | Sludge bld 4        | ABV 6/air line fitting                            | Small  | 3   |
| A4  | lime prep           | bad ABV/blowing air block valve                   | Large  | 10  |
| A5  | Scrubber 2 2nd flr  | ABV 545/E module fitting                          | Medium | 4   |
| A6  | Scrubber 2 2nd flr  | ABV 546/F module fitting                          | Medium | 4   |
| A7  | Scrubber 2 2nd flr  | ABV/A module blowingAir BV                        | Large  | 15  |
| A8  | Scrubber 2 1st flr  | W sump pp disch valve/Leaking from cyld green ABV | Large  | 15  |
| A9  | Scrubber2 2         | pp dich line flush ABV/3D ABV bad hose fitting    | Medium | 5   |
| A10 | Scrubber 1 4th flr  | Air BV/by elevator                                | Medium | 5   |
| A11 | Scrubber 1 3rd flr  | Quench vlv cyld/F module                          | Medium | 5   |
| A12 | Scrubber 1 4th flr  | Bad ABV/NE stair bad ABV                          | Large  | 10  |
| A13 | Scrubber 1 C module | Mist Elim ABV 195/2 CCC air leak around diaphram  | Large  | 10  |
| A14 | Scrubber 1 B Module | Temp. pacth leak/2nd flr ground level             | Large  | 10  |
| A15 | Scrubber 1 1st flr  | Cracked pipe ABV/2F SP PP F module                | Large  | 10  |
|     |                     |   | Total: | 855 |

## 4.5 POTENTIALLY INAPPROPRIATE USES OF COMPRESSED AIR

Potentially inappropriate uses of compressed air are demand-side applications that may be more efficiently handled by another power source rather than compressed air. The following sections identify and evaluate some inappropriate applications present at many plants.

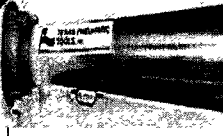
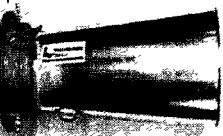
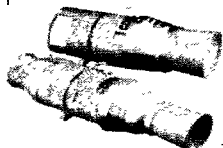


### 4.5.1 Air Movers or Air Horns

These items are part of a family of products known as "Portable Ventilators." They are available in various designs to move large volumes of air (1,000 to 10,000 cfm) in plants for many applications. The most common drives are electric, but they also come in Venturi air drives, which use high pressure compressed air to pull ambient outside air by a Venturi action. Generally, these use from 100 scfm each to 300 scfm for the most common 6" and 8" sizes.

The extensive use of air horns throughout the plant during the hot summer months (7 months). Can create a significant continuing load demand not accounted for in this audit.

The chart below lists the most popular 6" and 8" air horns and their annual electrical energy operating cost.

**Operating Cost Comparison for Air Horns**

|   | Model / Class               | Air Flow to Process | Compressed Air Usage @ 60 psig | Energy Cost        | Electric HP | Energy Cost     | Estimated Price per Unit |
|---|-----------------------------|---------------------|--------------------------------|--------------------|-------------|-----------------|--------------------------|
|   | TX6AM Compressed Air        | 2,885 cfm           | 98 (24 kW)                     | <b>\$6,000/yr</b>  |             |                 | \$400 - \$600            |
|   | TX8AM Compressed Air        | 4,152 cfm           | 152 (41.4 kW)                  | <b>\$10,350/yr</b> |             |                 | \$400 - \$600            |
|   | VANO 250 Electric           | 3,000 cfm           |                                |                    | 1 HP        | <b>\$200/yr</b> | \$2,000                  |
|  | Double Heat Killer Electric | 9,500 cfm           |                                |                    | 1 HP        | <b>\$200/yr</b> | \$3,000                  |
|  | TA 16 Electric              | 5,500 cfm           |                                |                    | 2 HP        | <b>\$400/yr</b> | \$2,600                  |

Operating cost based on \$0.05 /kWh – 5,000 hours/year.

- The compressed air-driven horns have a significant lower initial cost.
- The electrical energy cost savings of the electric-driven alternatives creates a very quick, simple payback.
- The air flows shown are at 80 psig; at 100 psig, they would be 20% higher.
- Vortec fixed flow vortex coolers are also often used to spot cool bearings. The approximate performance characteristics are:

| <b>Standard Model</b>      |         |         |          |
|----------------------------|---------|---------|----------|
| Size                       | 1"      | 2"      | 3"       |
| Compressed Air @ 80 psig   | 50 scfm | 75 scfm | 100 scfm |
| Temperature Drop / 80 psig | 60-80°F | 60-80°F | 60-80°F  |

The most popular size is the 100 cfm cooler. As long as these are used for spot cooling at or on a bearing, etc., for a limited time, they may be the best choice for effectiveness. If these can be replaced with a Vano electric-operated coolers, the savings would be about 100 cfm each (16.6 kW or \$4,050 per year in basic electric cost. These are significantly more expensive or less efficient than the standard air horns. We recommend that these be changed as fast as possible.

Reviewing the electric-driven air movers available from Coppus (and others), the following units would be appropriate substitutes for air horns to be used for cooling and ventilation and offer significantly savings.

- First choice for directed air flow such as cooling:

VANO 175 CV or 250 CV – these are powered by electric motor-driven axial vane fans, capable of large volume flow through ducting as required. They are available with totally enclosed motors (or explosion proof). They produce flow from 1,500 to 3,000 scfm and range from ½ hp to 1 hp.

Estimated cost for enclosed motor -- \$1,500 to \$2,000 each

- First choice for more drive and large runs of ducting:

TA 16 – tube axial blower with heavy-duty housing and “non-sparking” cast aluminum fan blades. They are also available with totally enclosed or explosion-proof motors. They flow 5,500 scfm of high drive air with a 2-hp electric motor.

Estimated cost for enclosed motor -- \$2,600 each.

- First choice for maximum cooling effect (larger area):

Double-duty Heat Killer – axial vane, electric-driven blower with adjustable guide vanes to lower or increase the velocity and change the flow patterns. These are available with enclosed or explosion-proof motors. Designed for high performance cooling and effective on “air heat quenching.” The recommended size would be the portable 24” K10 with a 1-hp electric motor moving 9,500 cfm.

These can also be equipped with an optional "Cold Front" evaporative cooler, "to lower the temperature of the cooling air." It is our opinion that this will not be needed.

Estimated cost with enclosed motor is \$3,000 each.

For such applications as furnace cooling, you may find combinations of some of the above will be more effective. Certain units can be run as primary ventilators and later used for direct cooling.

- ☒ **RECOMMENDED PROJECT (#8)** – Remove all existing air horns from use and replace them with the appropriate number of electric units. Utilize the electric horns and train personnel in their use.

|  |            |
|--|------------|
| Equivalent number of air-operated air horns operating  | 10 units   |
| Operating hours (summer season only)   | 2190 hours |
| Total operating cost: 10 air horns x 124 scfm each x 1 kW per scfm x<br>2190 hours in summer season x 5¢ per kWh | \$27,156   |
| Total operating cost of electric-operated air horns (negligible or < \$1000)                                     | \$0        |
| Equipment cost (10 units)  | \$20,000   |

#### 4.5.2 Air-Operated Diaphragm Pumps

Although air-operated diaphragm pumps are not very energy efficient, they tolerate aggressive conditions relatively well and run without catastrophic damage even if the pump is dry. There are several questions to ask and areas to investigate that may yield significant air savings:

- Is an air-operated diaphragm pump the right answer? An electric pump is significantly more energy efficient. Electric motor driven diaphragm pumps are readily available. An electric motor drive progressive cavity pump may also well work.
- Consider installing electronic or ultrasonic controls to shut pumps off automatically when not needed. Remember that pumps waste the most air when they are pumping nothing.
- Is the pump running most of the time at the lowest possible pressure? The higher the pressure is, the more air is used. For example, filter-packing operations often do not need high pressure except during the final stages of the filter packing cycle. Controls can be arranged to generate lower pressures in the early stages and higher pressures later on, which may generate significant savings.

### Air-Operated Diaphragm Pumps

| Nominal Pump Size | Air Pressure Range                  | Nominal Scfm | Displacement (gals/cycle) | Estimated Electric Pump HP |
|-------------------|-------------------------------------|--------------|---------------------------|----------------------------|
| 1"                | 65 - 100 psig<br>(average 80 psig)  | 25 – 30      | 0.08                      | ¾ hp – 1 hp                |
| < ½"              | 65 - 100 psig<br>(average 20 psig)  | 15 – 20      | 0.09                      | ¾ hp – 1 hp                |
| 1 ½"              | 70 – 100 psig<br>(average 80 psig)  | 45 – 58      | 0.34                      | 1 ½ hp – 2 ½ hp            |
| 2"                | 80 – 120 psig<br>(average 95 psig)  | 90 – 120     | 0.43                      | 3 hp – 5 hp                |
| 3"                | 90 – 100 psig<br>(average 100 psig) | 125 – 150    | 1.25                      | 5 hp – 7 hp                |

- The above numbers are based on pumping water
- Flow varies by brand, model, and application
- Pressure requirement varies by brand, model, and application
- Air flow goes up as the flow and pumping cycles per minute increase
- Pressure requirement (air) may rise as head increases
- The electric pump horsepower will increase significantly at higher head.

All of the diaphragm pumps were smaller than 1 ½" and were run well controlled only as needed.

- ☒ **PHASE 2 RECOMMENDATION** – Review all air-operated diaphragm pump operating costs with those of an appropriate electric unit.

#### 4.5.3 Air Motors and Hoists

Compressed air is a very inefficient transfer of energy requiring 7 to 8 hp of electrical energy to produce enough air to deliver 1 hp worth of work. For this reason, air hoists and air motors are often good targets to be replaced by electric-powered units. These applications use 15 to 20 cfm each or the equivalent of 5 hp worth of air.

Air hoists are rated in tonnage of capacity. Often the air motor horsepower is the same for several different tonnage ratings. Care should be taken to review the actual performance chart of the hoist in question.

- ☒ **PHASE 2 RECOMMENDATION** – Continue to look for air motor/hoist applications. Monitor incoming equipment.

#### 4.5.4 Air Vibrators

Air vibrators are used to keep product or packaging moving or separated – e.g., keeping lids separated prior to sealing. If a plant employs air vibrators that use about 10 cfm each, they will require about 2.5 hp or more to produce the same as a similar electric vibrator, which might use about 0.25-hp input energy.

- ☒ **PHASE 2 RECOMMENDATION** – Continue to look for air air-operated vibrator applications. Monitor incoming equipment.

Date:

Feb 2008

AIR POWER

Compressed Air System Audit  
Data Acquisition FormInter Mountain Power  
COMPANY NAME850 W. Baugh Wellman Rd  
STREETDelta UTAH 84629-9522  
CITY STATE ZIP435-864-6810 435-864-6865  
TELEPHONE FAX NUMBERCompress Pump # Set 801-973 - 0154  
Kevin Sullivan c-801-633-7323  
PRIMARY CONTACT / TITLEWill Lovell Ass Super of maint-  
SECONDARY CONTACTPower Plant  
TYPE OF PLANTBret Kent / Richard Schmit  
E-MAIL ADDRESS

Nancy Bennett (Buyer)

## CURRENT EQUIPMENT DATA

Location: Main Compressor Room Ground Floor

## Unit 1: Compressor

1a-1b-

Brand Elliott Type BOV  
Model 310 DAB SN#  
HP(bhp) 700 Drive  
Cfm 2245 Psig 125Aftercooler  
Capacity control type  
Hours of operation per year

## Unit 2: Compressor

1b

Brand Elliott Type BOV  
Model 310 DAB SN#  
HP(bhp) 700 Drive  
Cfm 2245 Psig 125Aftercooler  
Capacity control type  
Hours of operation per year

## Unit 3: Compressor

1c

Brand Elliott Type BOV  
Model 310 DAB SN#  
HP(bhp) 700 Drive  
Cfm 2245 Psig 125Aftercooler w/c  
Capacity control type mod  
Hours of operation per year

## Unit 4: Compressor

1d

Brand Elliott Type  
Model 70 310 DAB SN#  
HP(bhp) 700 Drive  
Cfm 2245 Psig 125Aftercooler  
Capacity control type  
Hours of operation per yearCompressor discharge pressure Minimum 100 psig Maximum 125 psig  
System data: Min operating pressure Max operating pressureAverage cfm demand 1<sup>st</sup> shift cfm hrs/day days/yr  
Average cfm demand 2<sup>nd</sup> shift cfm hrs/day days/yr  
Average cfm demand 3<sup>rd</sup> shift cfm hrs/day days/yr  
Weekend / holiday cfm hrs/day days/yrPower rate .05 /kWh: Is there a demand charge? If yes, explain Blended Cost  
Based on what they have to payPower Factor Rating .9 Voltage 6600 Phase 3 ph Hz 60 Hz

IP12\_006222



## CURRENT EQUIPMENT DATA

 Location: Dryer Room - 2nd Floor

## Unit 1: Dryer

A

 Brand SCAB Dryer Type \_\_\_\_\_  
 Model \_\_\_\_\_ SN# \_\_\_\_\_  
 HP(kW) \_\_\_\_\_ Regen \_\_\_\_\_  
 Cfm 2242 Psig 125  
 Blower/heater kW \_\_\_\_\_  
 Cooling type \_\_\_\_\_  
 Capacity control type \_\_\_\_\_  
 Percent purge \_\_\_\_\_

## Unit 5: Compressor

Ignister Compressor

2 cyl. Horizon  
Booster System
 Brand \_\_\_\_\_ Type \_\_\_\_\_  
 Model \_\_\_\_\_ SN# \_\_\_\_\_  
 HP(bhp) \_\_\_\_\_ Drive \_\_\_\_\_  
 Cfm \_\_\_\_\_ Psig \_\_\_\_\_

 Aftercooler \_\_\_\_\_  
 Capacity control type \_\_\_\_\_  
 Hours of operation per year \_\_\_\_\_

## Unit 2: Dryer

B

 Brand \_\_\_\_\_ Type \_\_\_\_\_  
 Model \_\_\_\_\_ SN# \_\_\_\_\_  
 HP(kW) \_\_\_\_\_ Regen \_\_\_\_\_  
 Cfm 2242 Psig \_\_\_\_\_  
 Blower/heater kW \_\_\_\_\_  
 Cooling type \_\_\_\_\_  
 Capacity control type \_\_\_\_\_  
 Percent purge \_\_\_\_\_

## Unit 6: Compressor

PHE style.  
Ignister Compressor
 Brand \_\_\_\_\_ Type \_\_\_\_\_  
 Model \_\_\_\_\_ SN# \_\_\_\_\_  
 HP(bhp) \_\_\_\_\_ Drive \_\_\_\_\_  
 Cfm \_\_\_\_\_ Psig \_\_\_\_\_

 Aftercooler \_\_\_\_\_  
 Capacity control type \_\_\_\_\_  
 Hours of operation per year \_\_\_\_\_

## Unit 3: Dryer

C

 Brand \_\_\_\_\_ Type \_\_\_\_\_  
 Model \_\_\_\_\_ SN# \_\_\_\_\_  
 HP(kW) \_\_\_\_\_ Regen \_\_\_\_\_  
 Cfm 2242 Psig \_\_\_\_\_  
 Blower/heater kW \_\_\_\_\_  
 Cooling type \_\_\_\_\_  
 Capacity control type \_\_\_\_\_  
 Percent purge \_\_\_\_\_

## Unit 7: Compressor

 Brand \_\_\_\_\_ Type \_\_\_\_\_  
 Model \_\_\_\_\_ SN# \_\_\_\_\_  
 HP(bhp) \_\_\_\_\_ Drive \_\_\_\_\_  
 Cfm \_\_\_\_\_ Psig \_\_\_\_\_

 Aftercooler \_\_\_\_\_  
 Capacity control type \_\_\_\_\_  
 Hours of operation per year \_\_\_\_\_

 2800 CFM AFTER Filter  
 2800 CFM Pre Filter

Location: \_\_\_\_\_

## Unit 4: Dryer

D

 Brand \_\_\_\_\_ Type \_\_\_\_\_  
 Model \_\_\_\_\_ SN# \_\_\_\_\_  
 HP(kW) \_\_\_\_\_ Regen \_\_\_\_\_  
 Cfm 2242 Psig 125  
 Blower/heater kW \_\_\_\_\_  
 Cooling type \_\_\_\_\_  
 Capacity control type \_\_\_\_\_  
 Percent purge \_\_\_\_\_

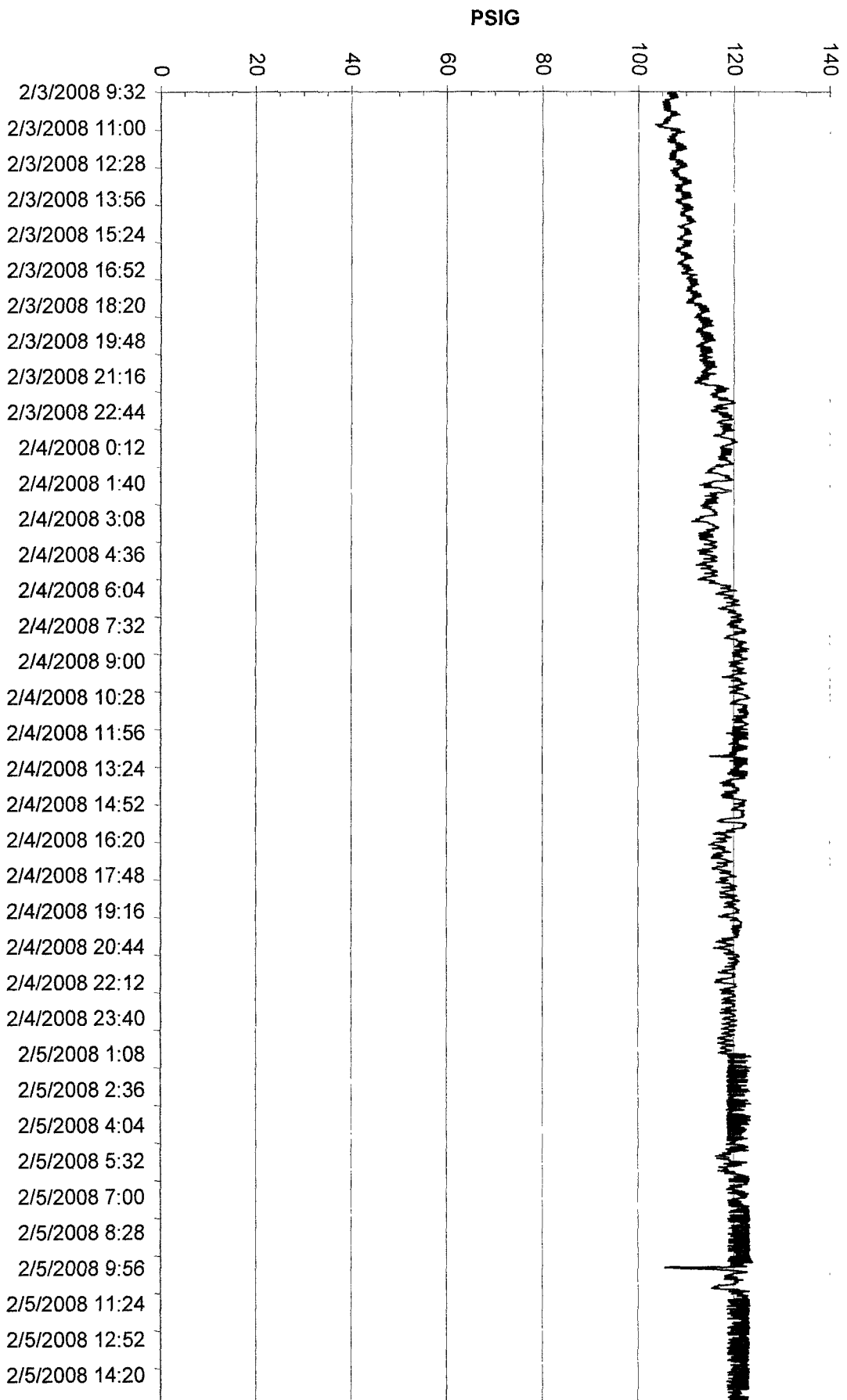
## Unit 8: Compressor

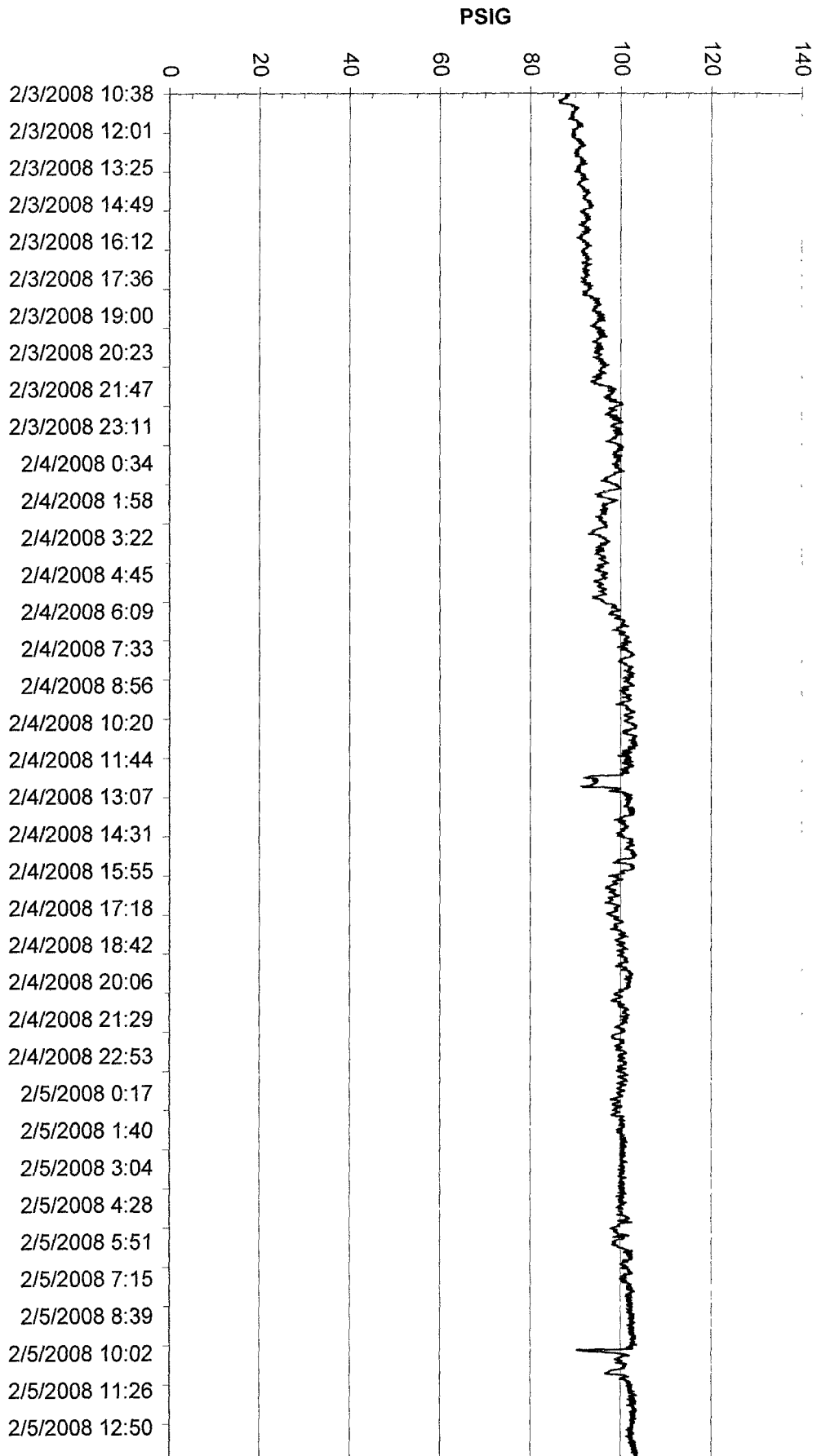
 Brand \_\_\_\_\_ Type \_\_\_\_\_  
 Model \_\_\_\_\_ SN# \_\_\_\_\_  
 HP(bhp) \_\_\_\_\_ Drive \_\_\_\_\_  
 Cfm \_\_\_\_\_ Psig \_\_\_\_\_

 Aftercooler \_\_\_\_\_  
 Capacity control type \_\_\_\_\_  
 Hours of operation per year \_\_\_\_\_

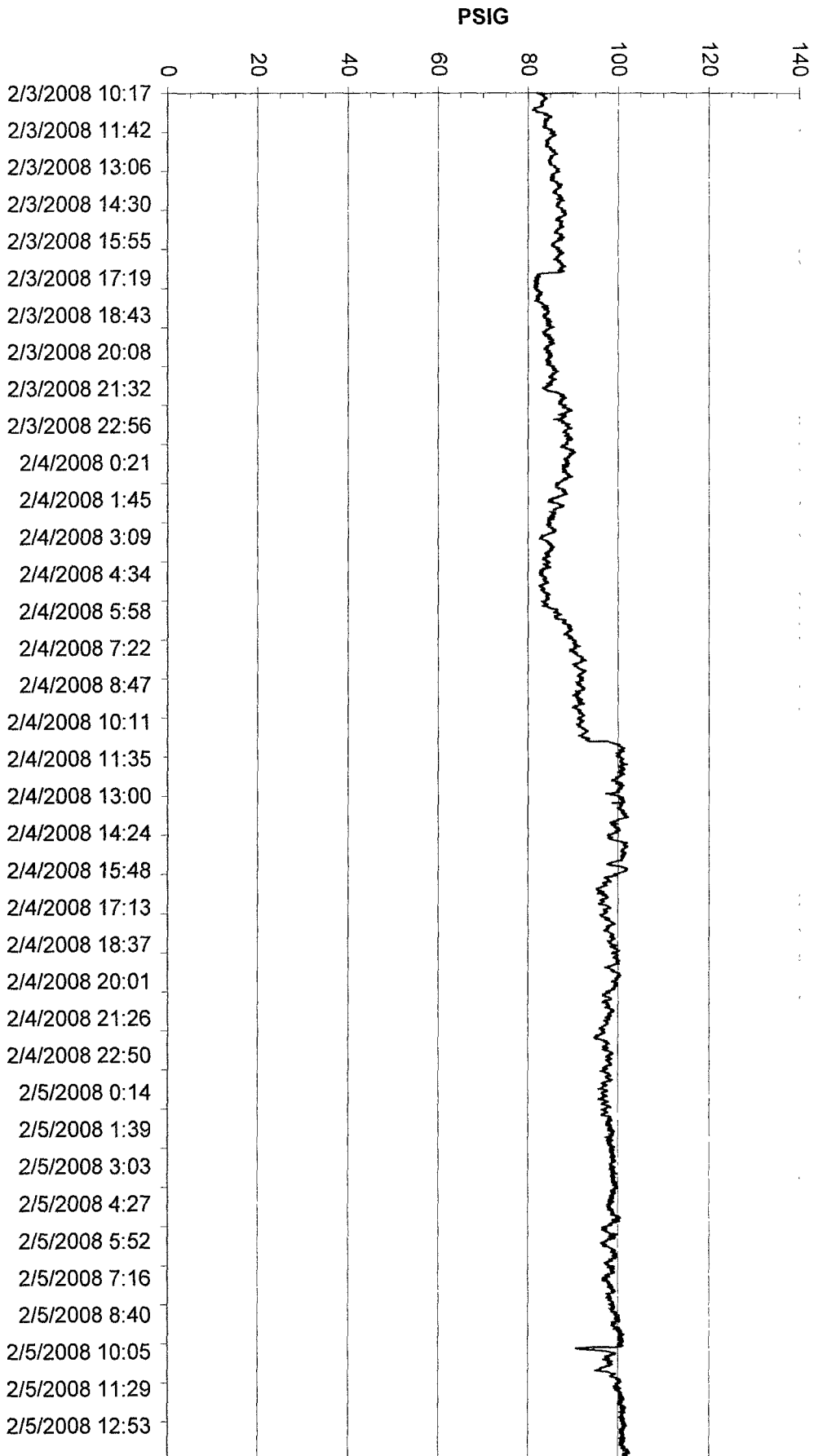
COMMENTS: \_\_\_\_\_

IPSC  
Compressor 1 Receiver

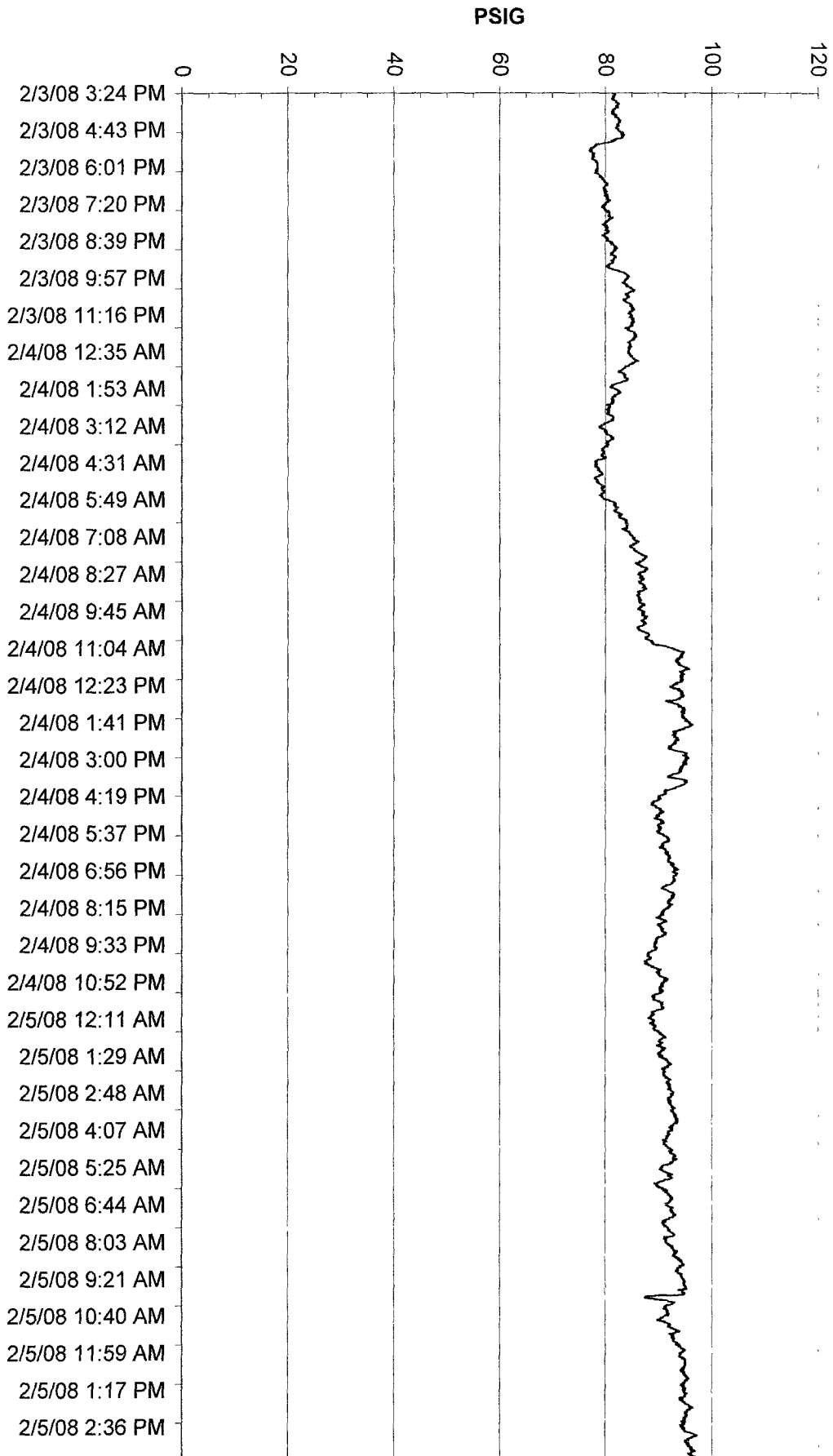




IPSC  
Unit 1 Scrubber  
Reciever

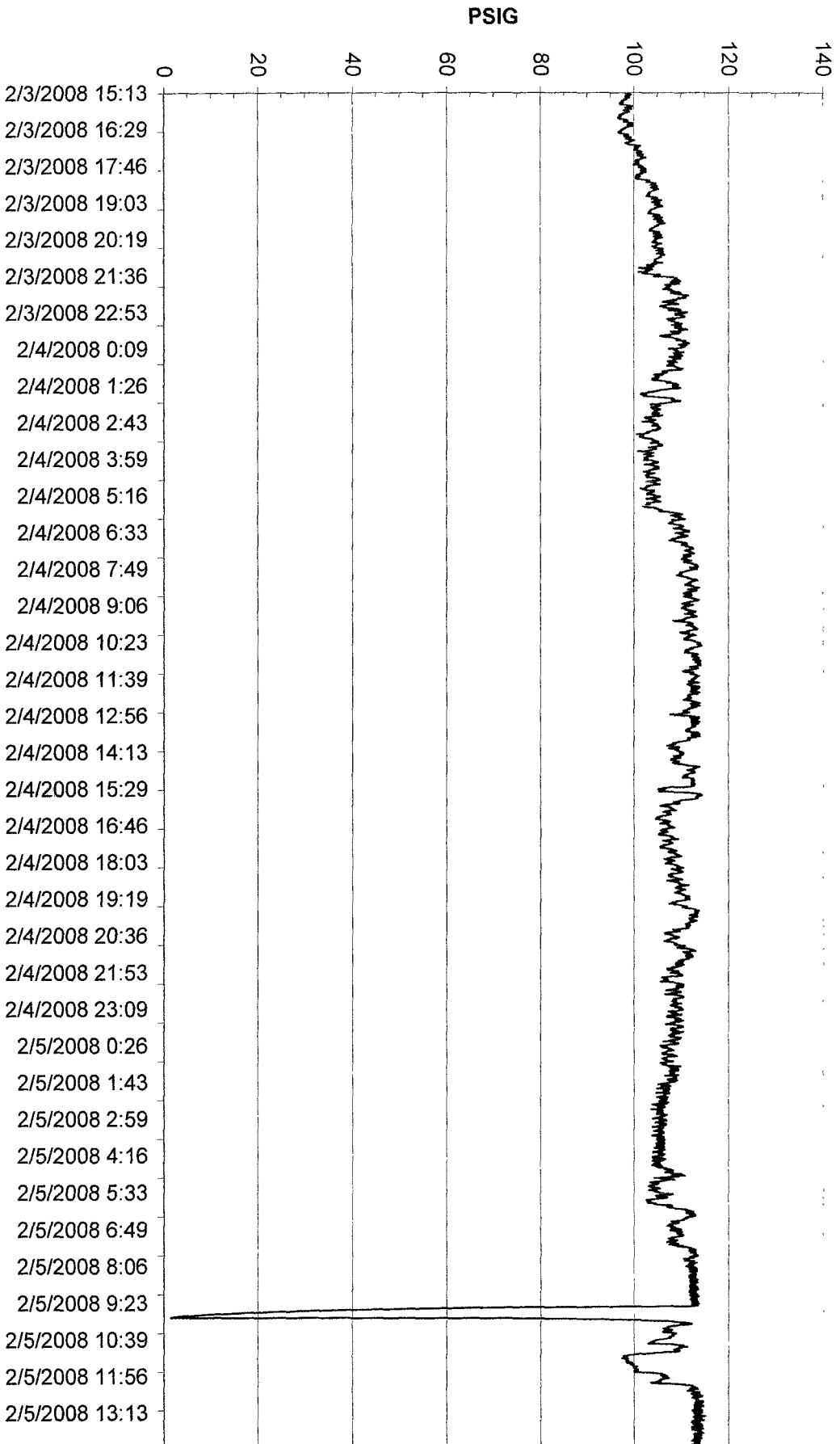


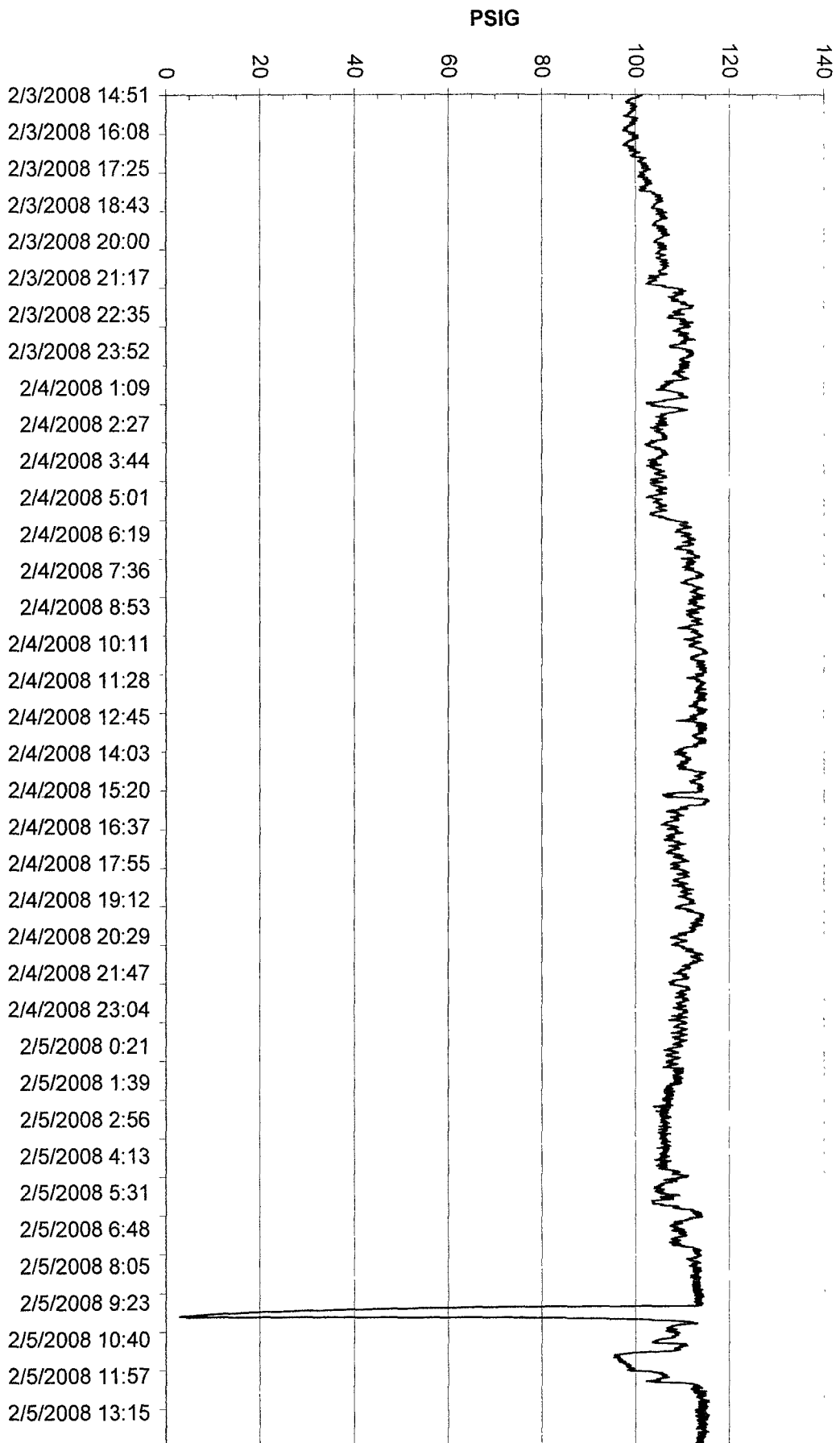
IPSC  
Unit 2 Scrubber  
Receiver



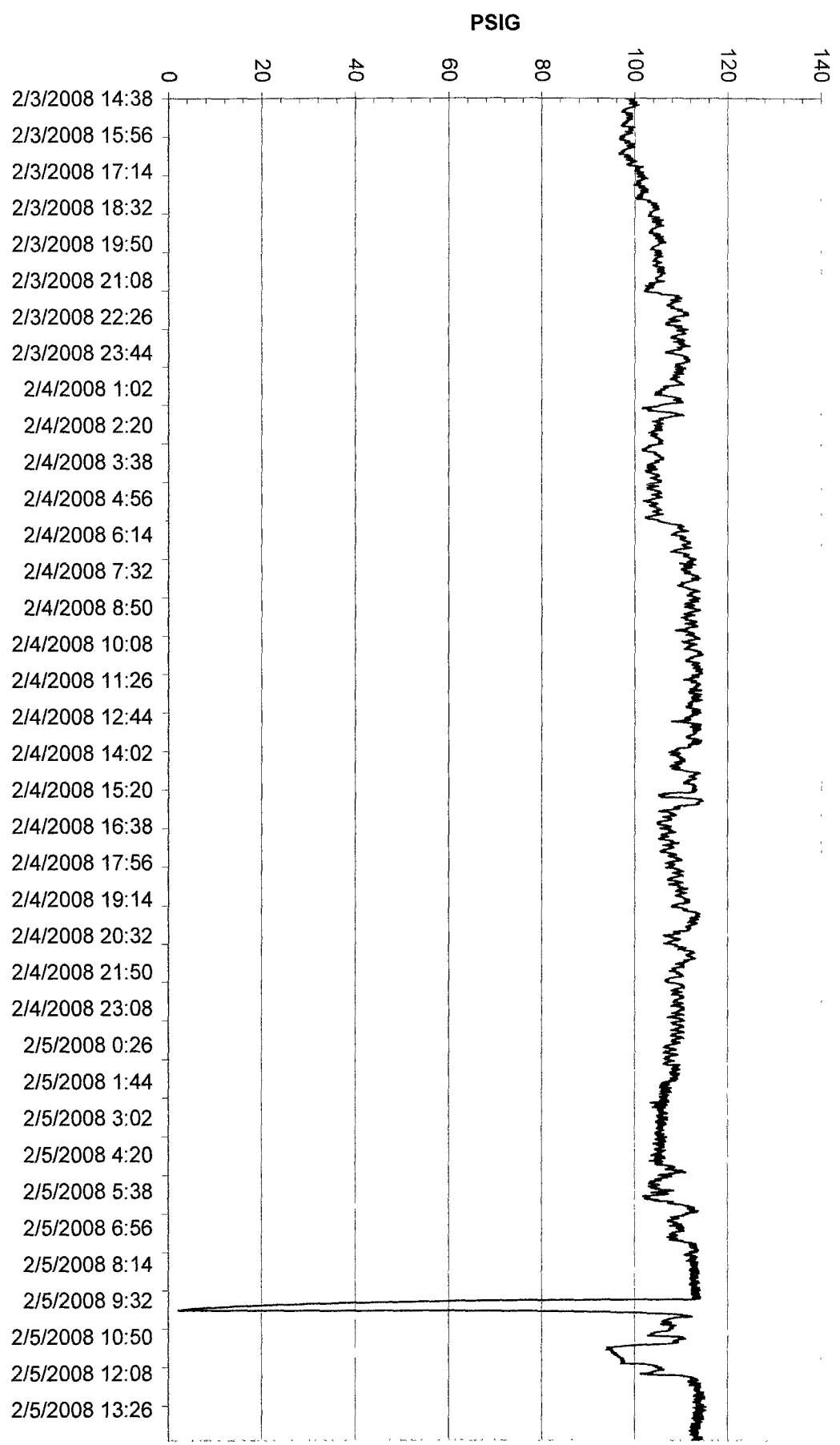
IPSC  
Unit 2 Tank C

IPSC  
Transfer Building 4



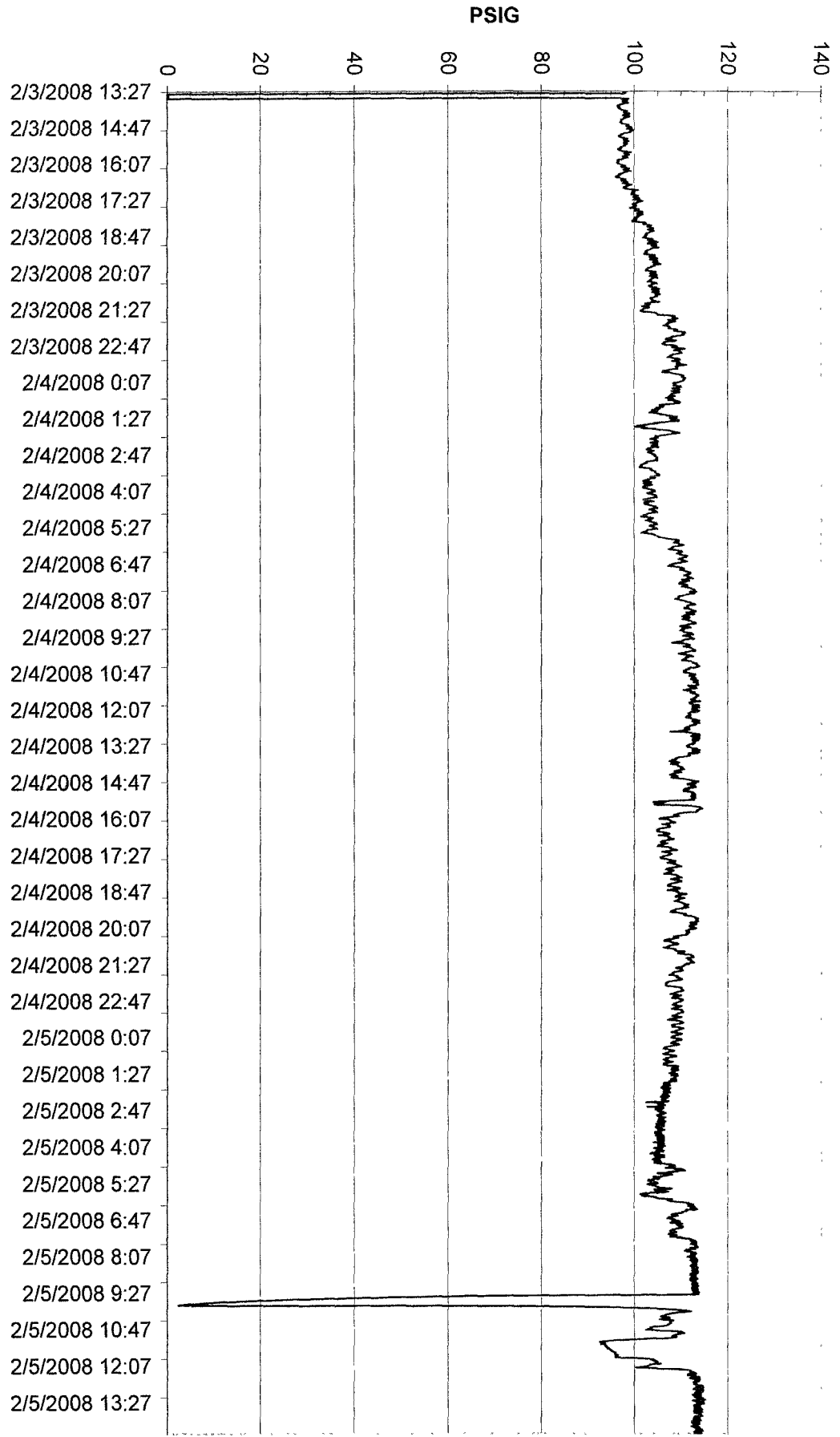


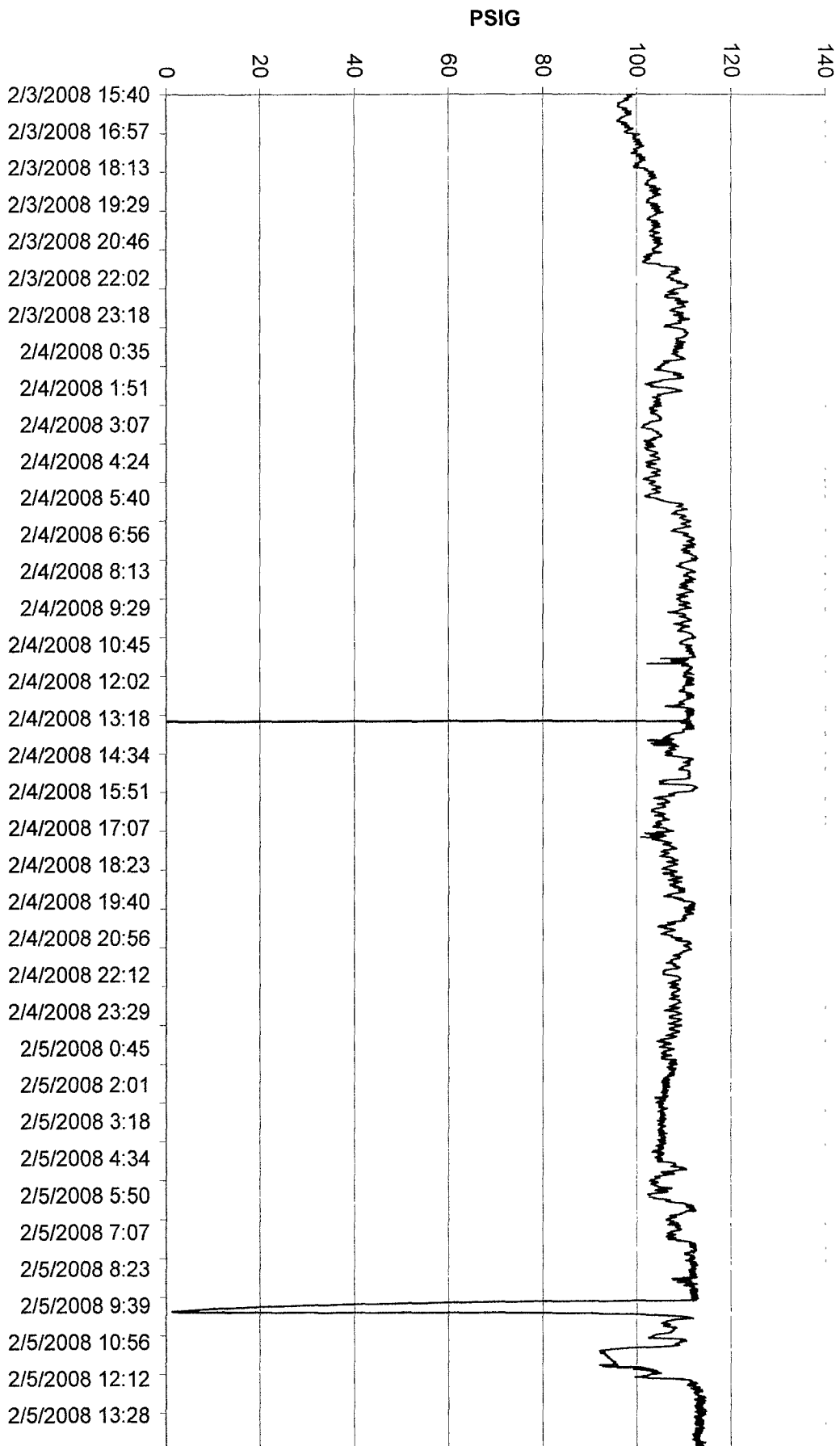
IPSC  
Transfer Building 2



IPSC  
Transfer Building 1

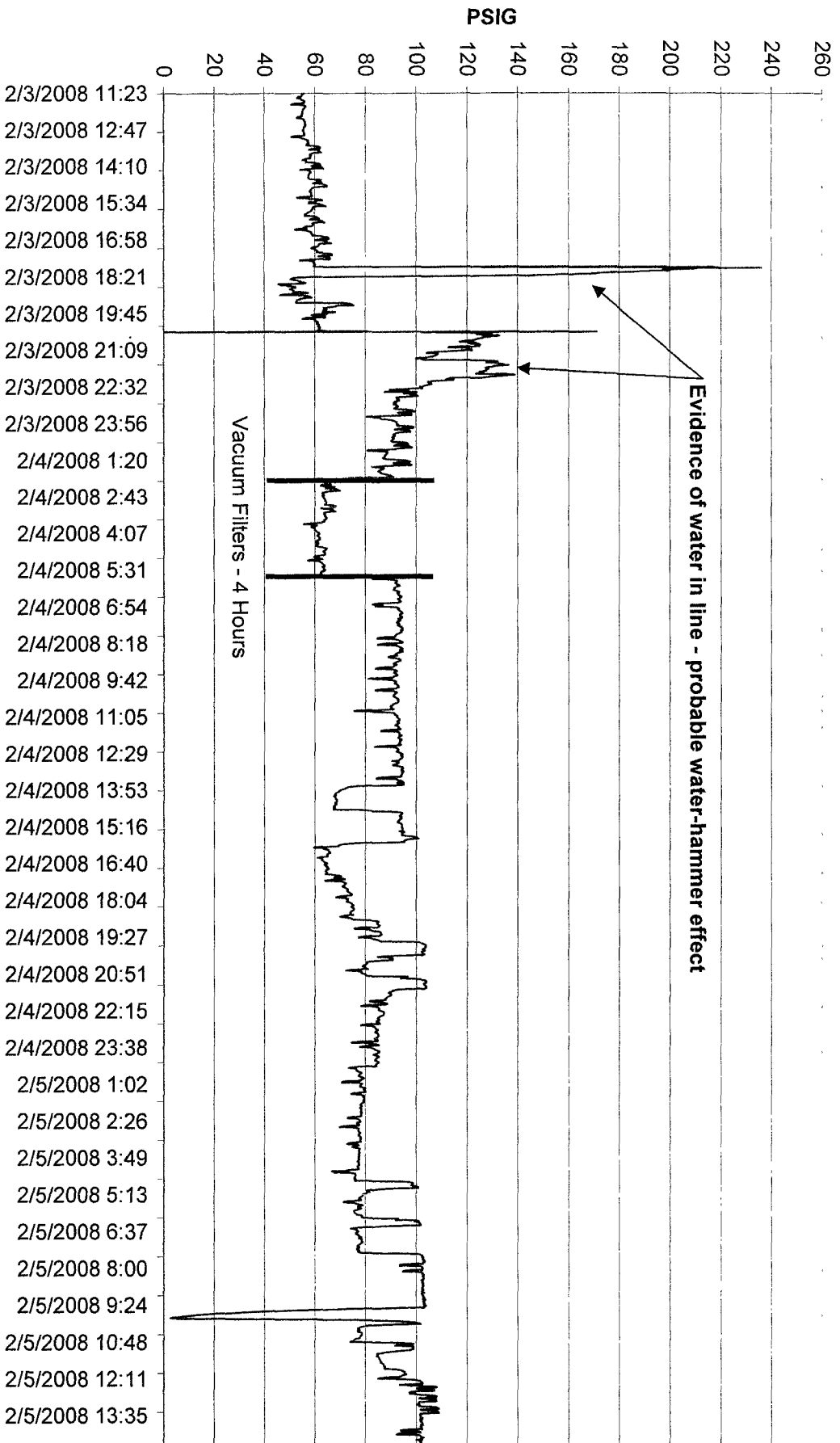
IPSC  
Coal Car Unload



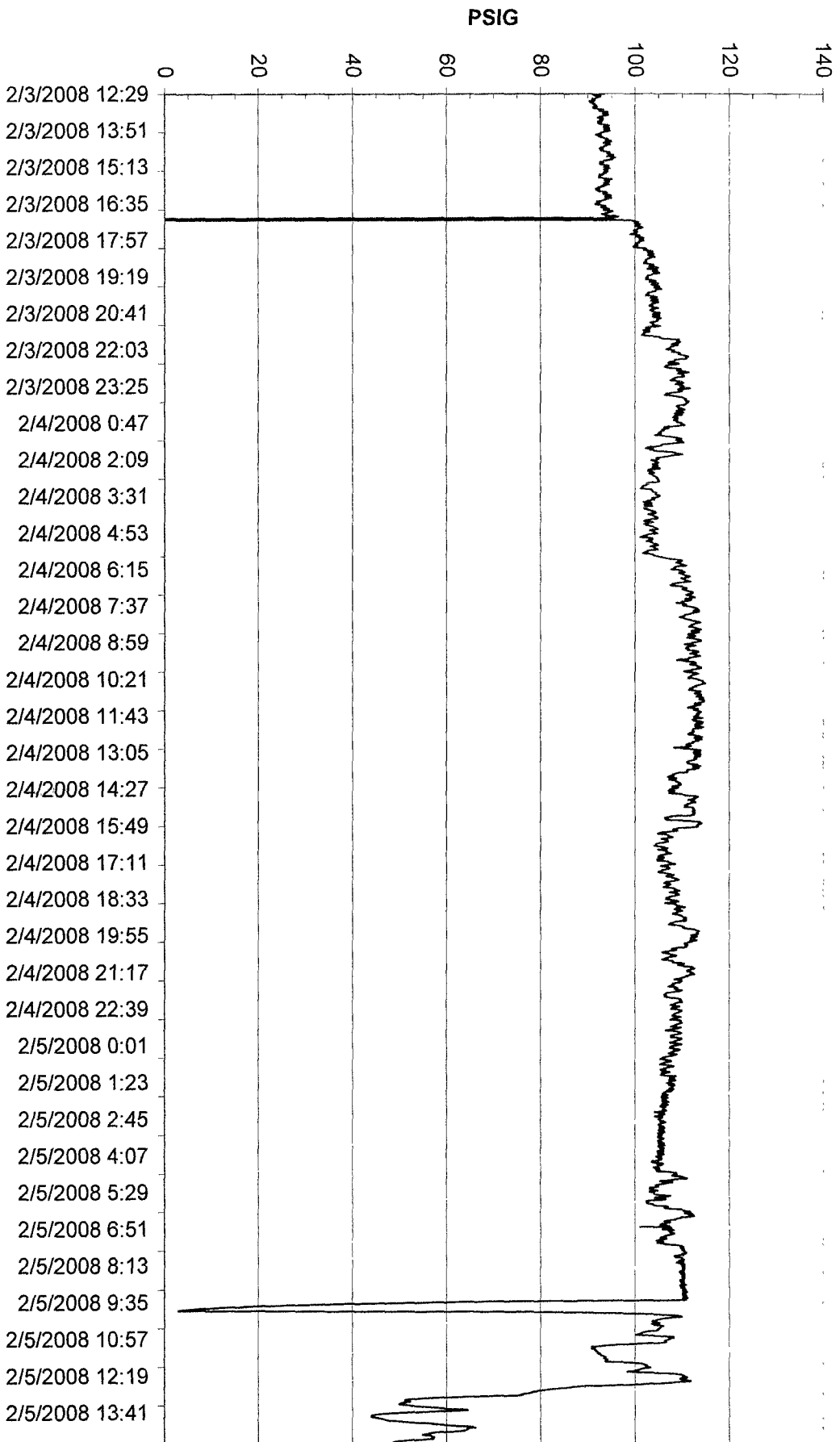


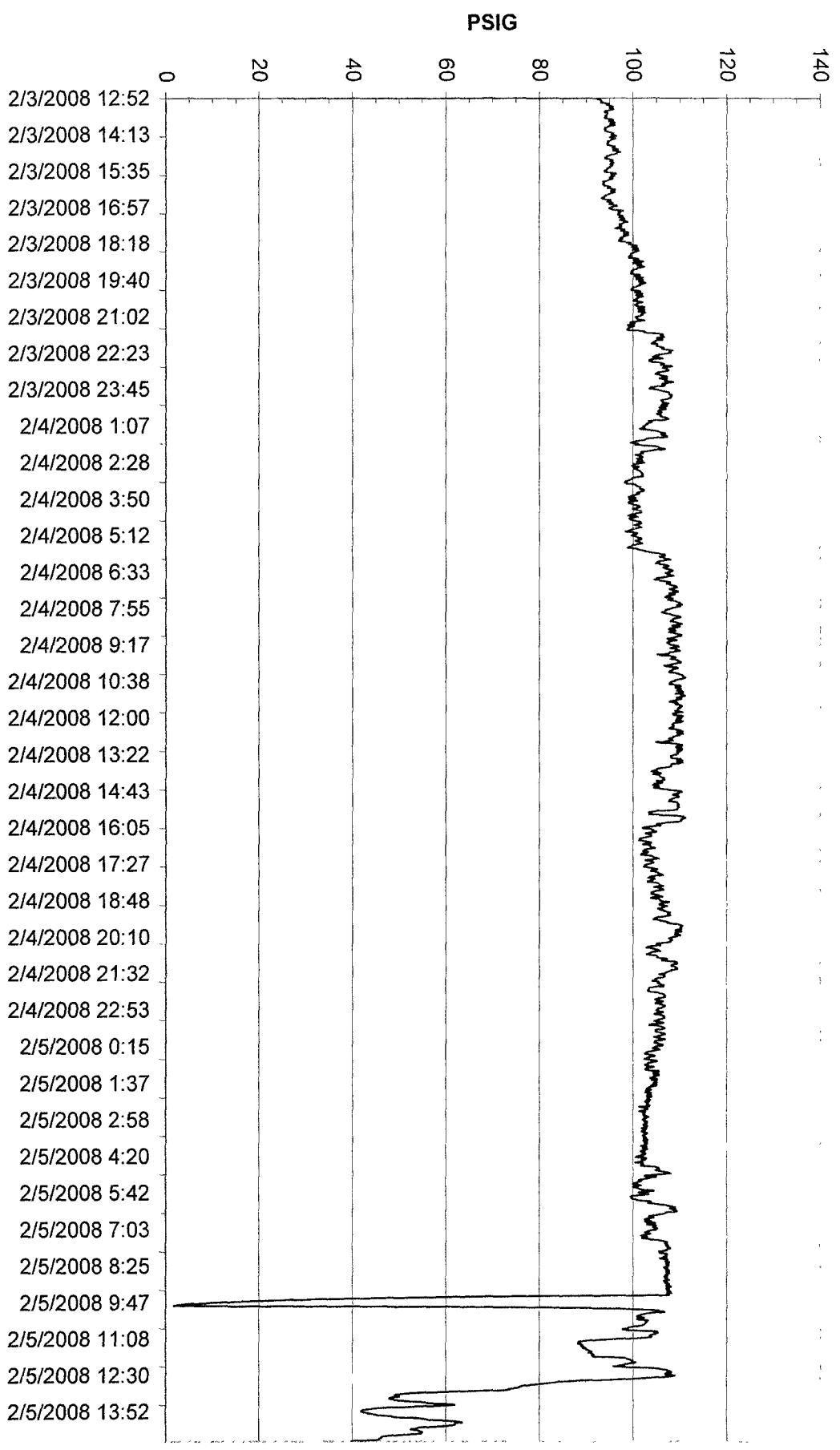
IPSC  
Limestone Unload

IPSC  
Sludge Transfer

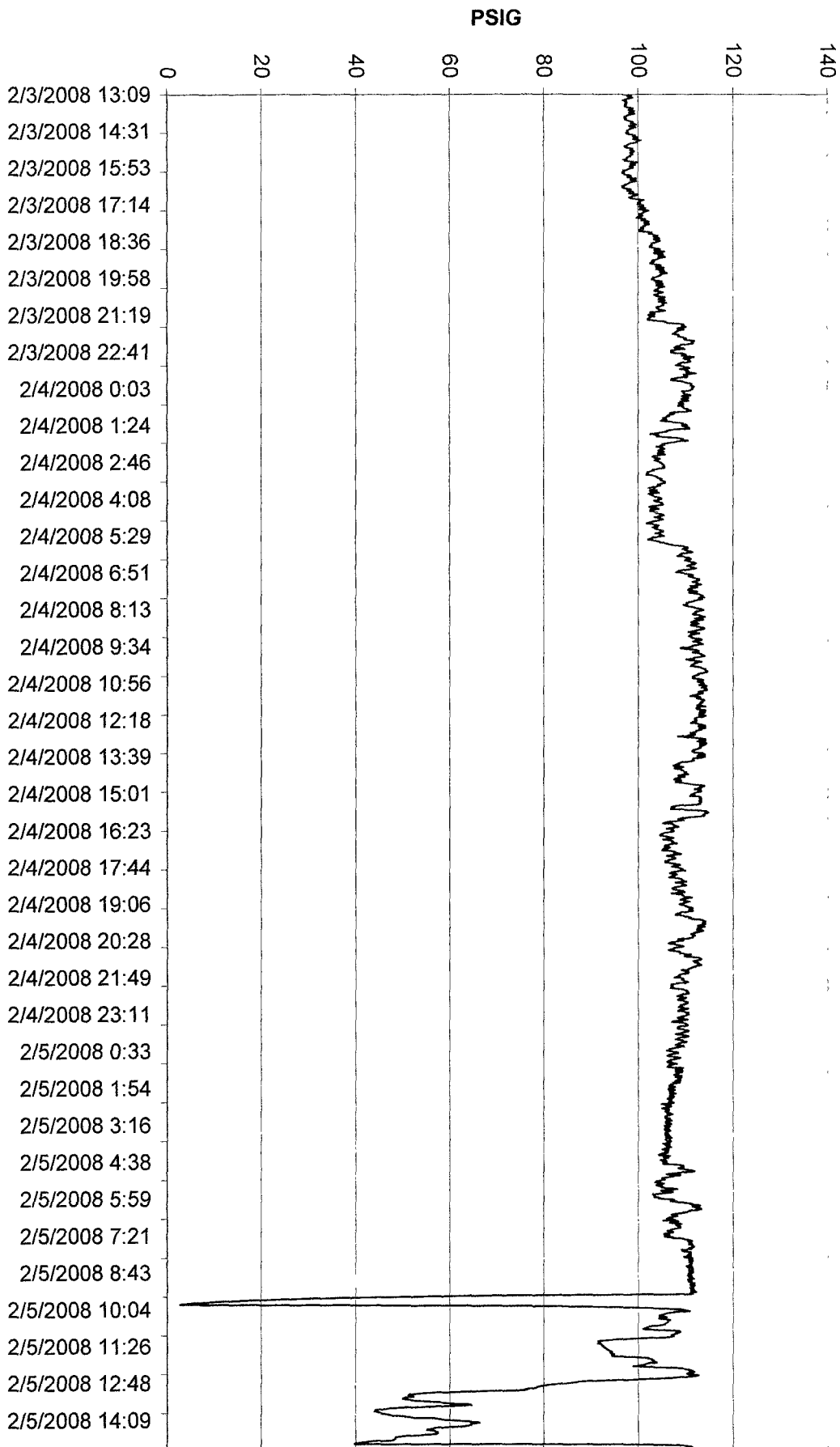


IPSC  
Stack

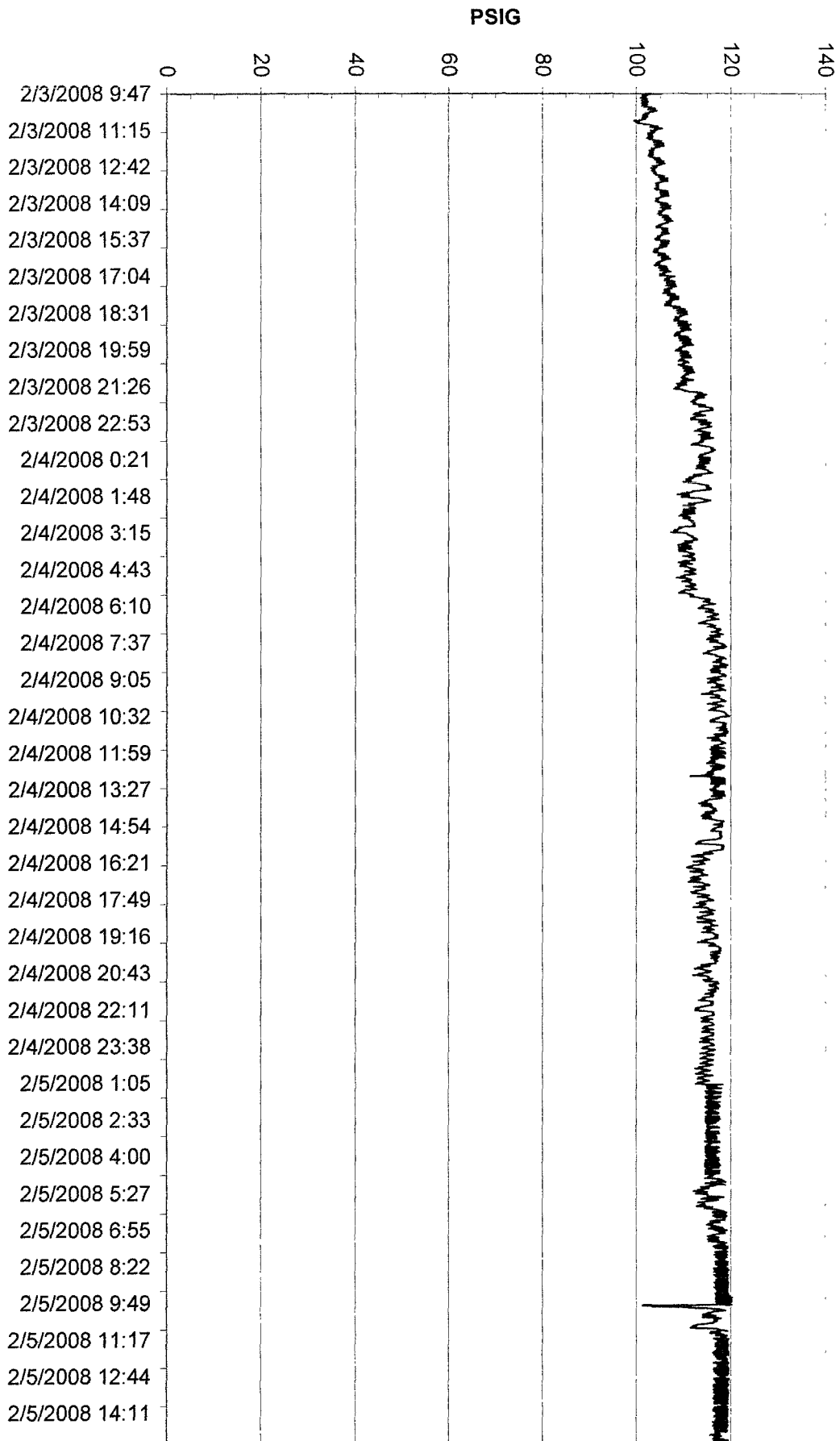




IPSC  
Water Treatment

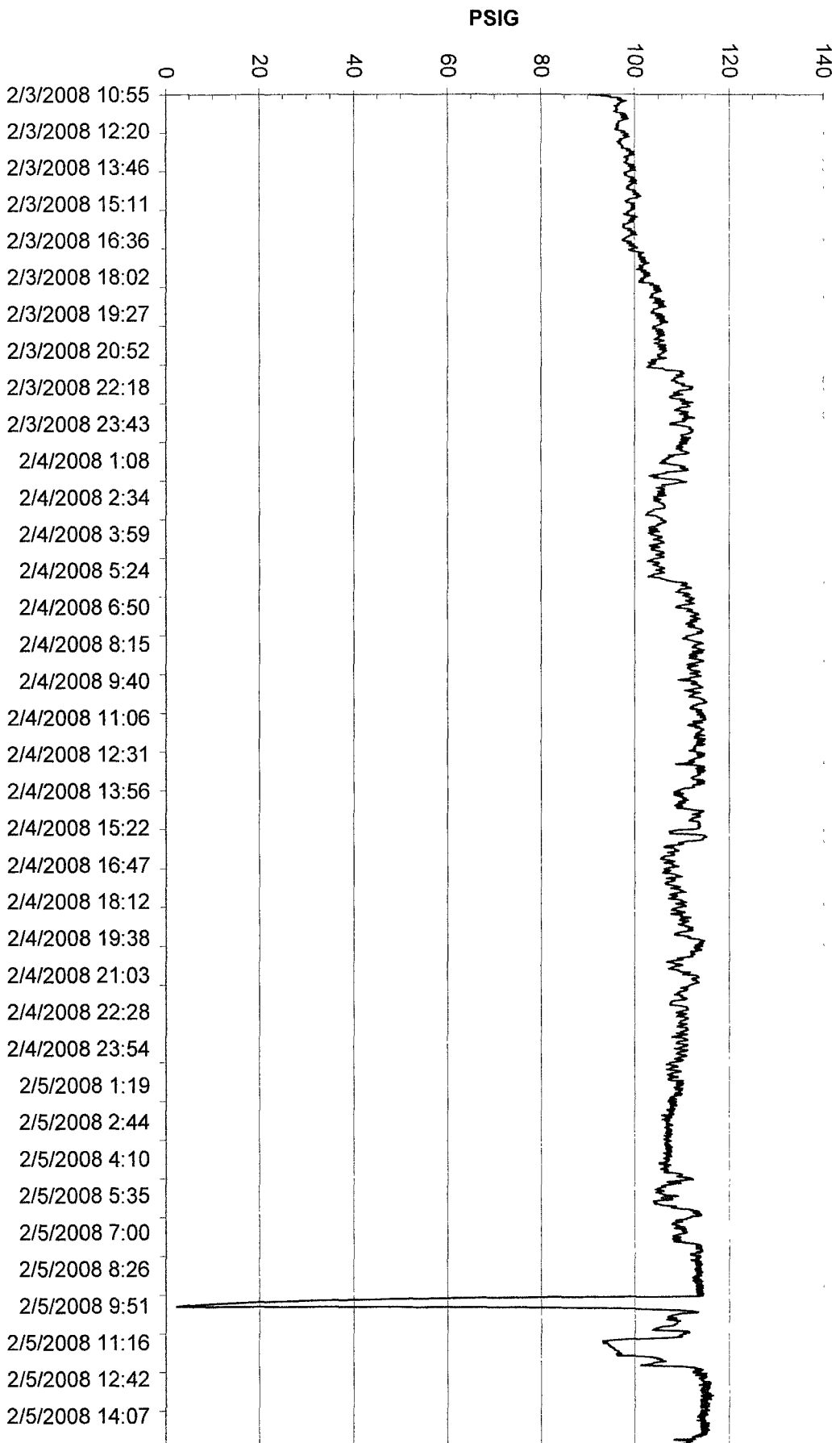


IPSC  
BLR Sump Transfer

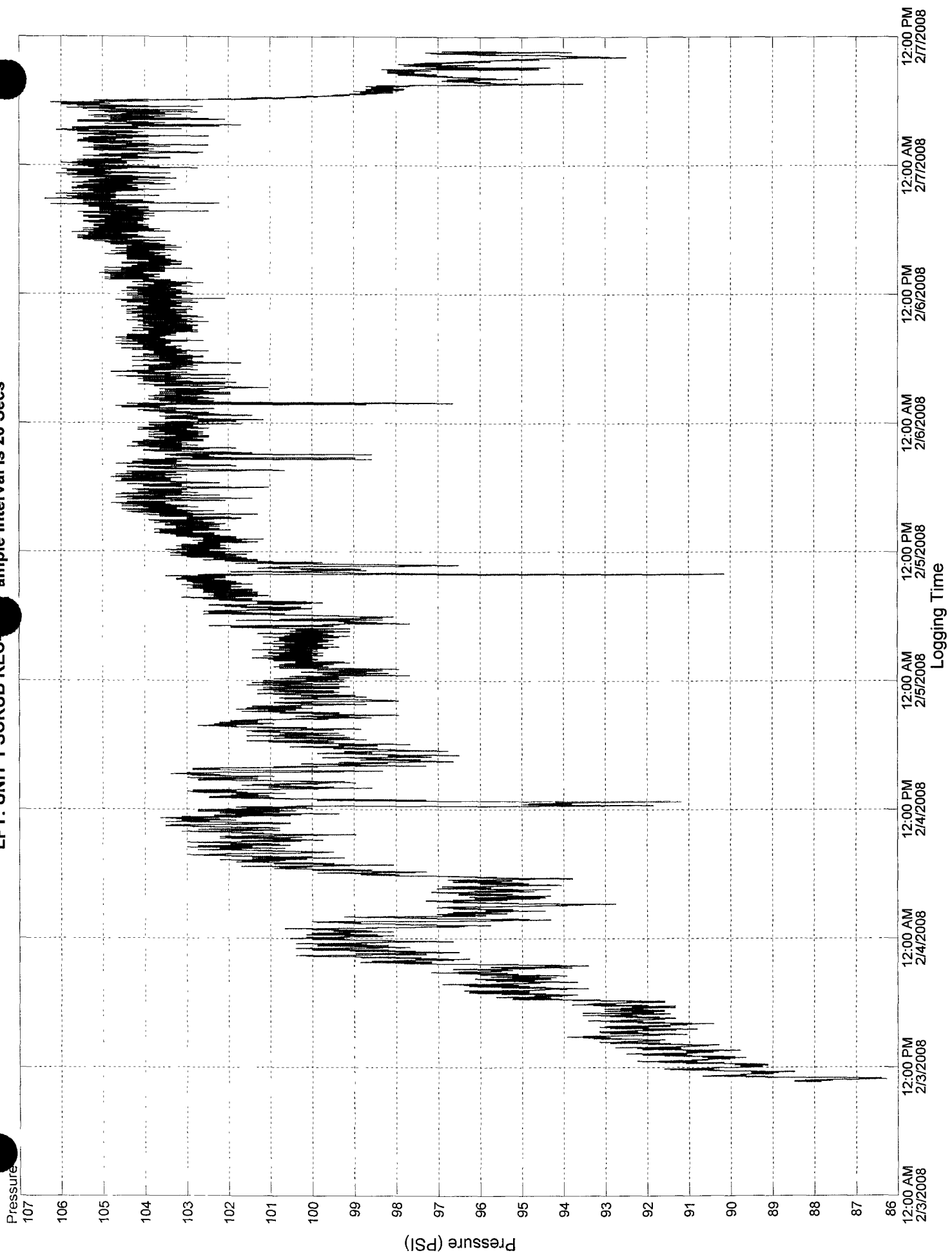


IPSC  
Ignition Boost

IPSC  
Lime Prep

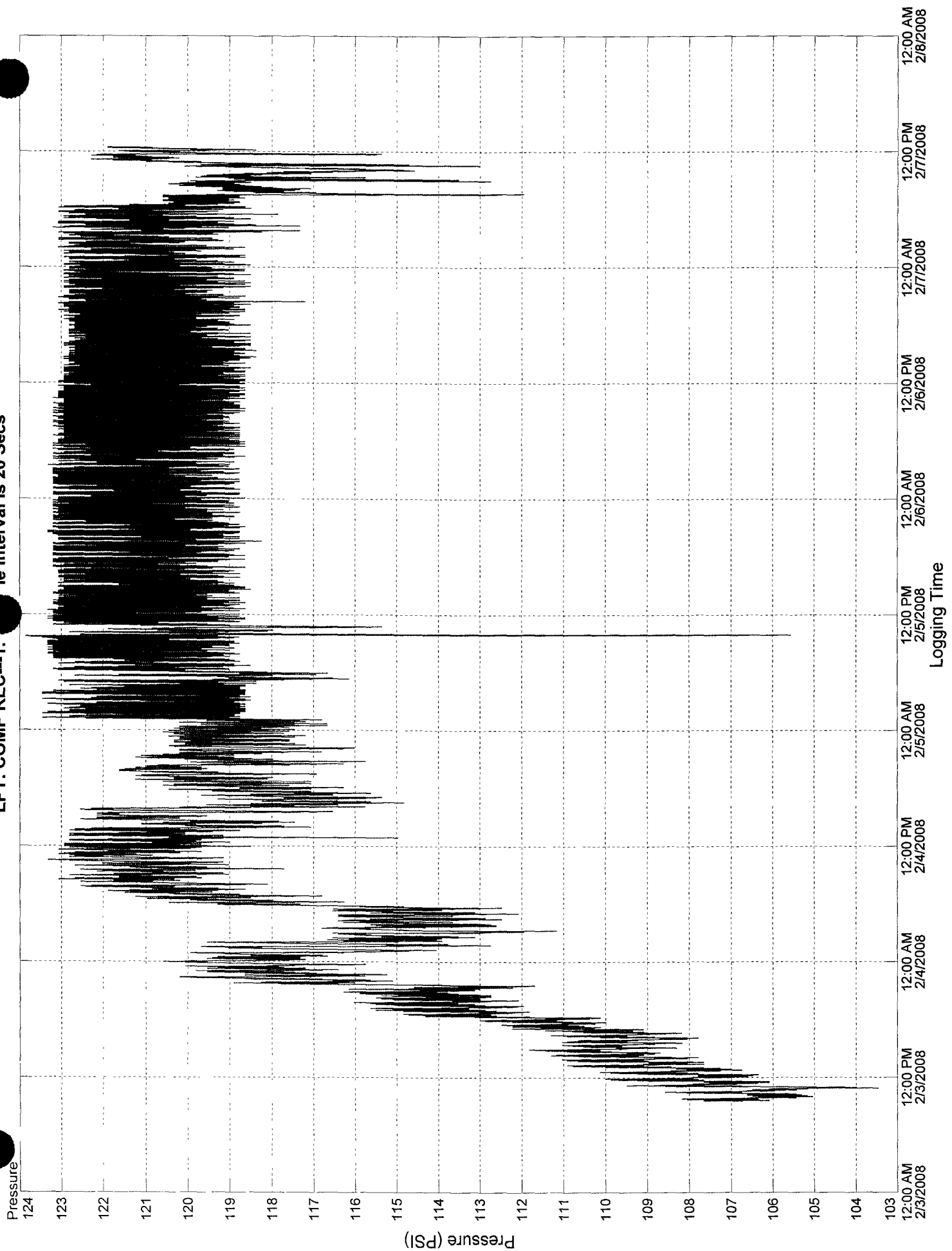


LPT: UNIT 1 SCRUB REC- sample Interval Is 20 Secs



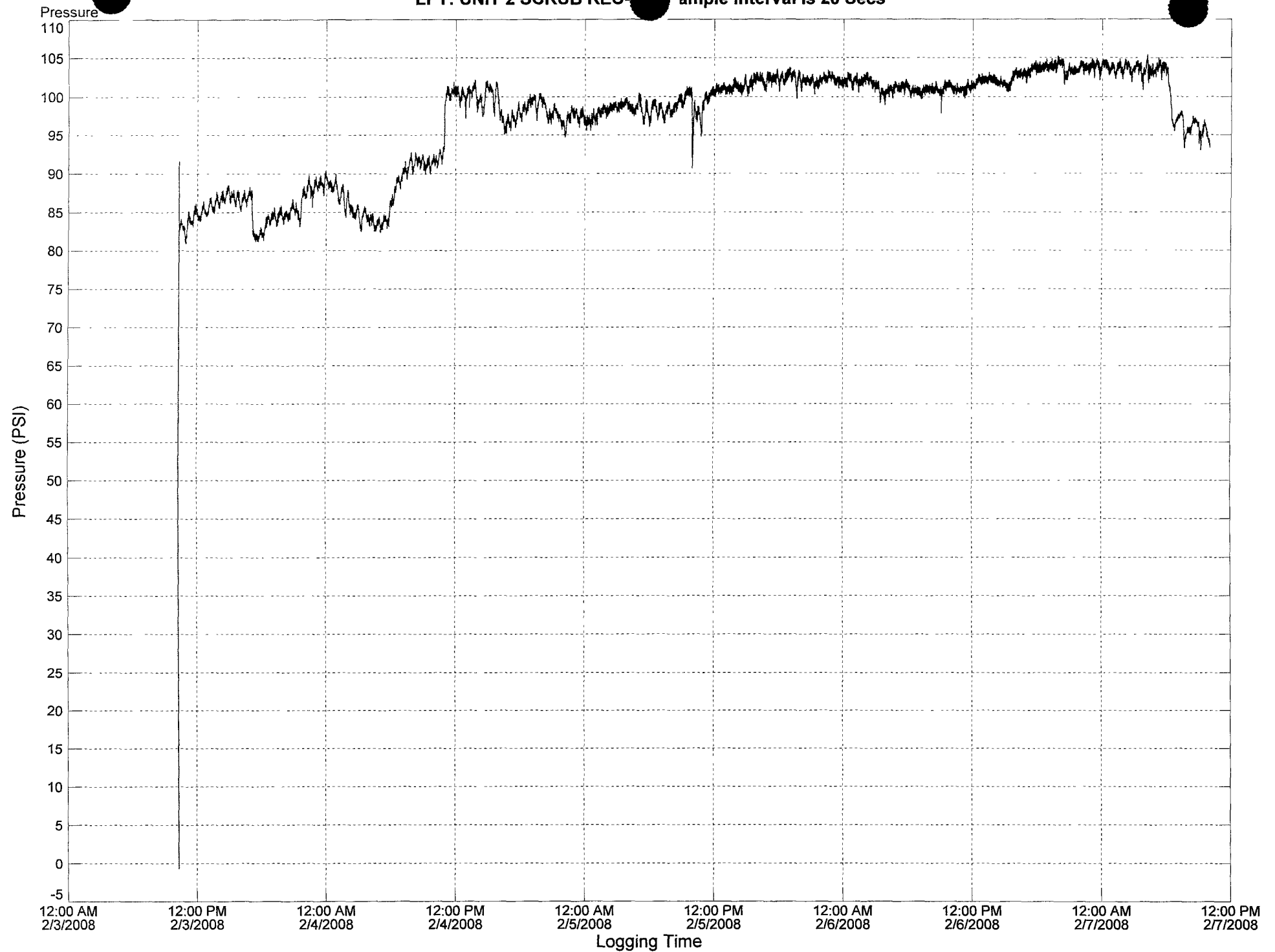
IP12\_006239

LPT: COMP REC---1. le Interval Is 20 Secs



IP12\_006240

LPT: UNIT 2 SCRUB REC ample Interval Is 20 Secs



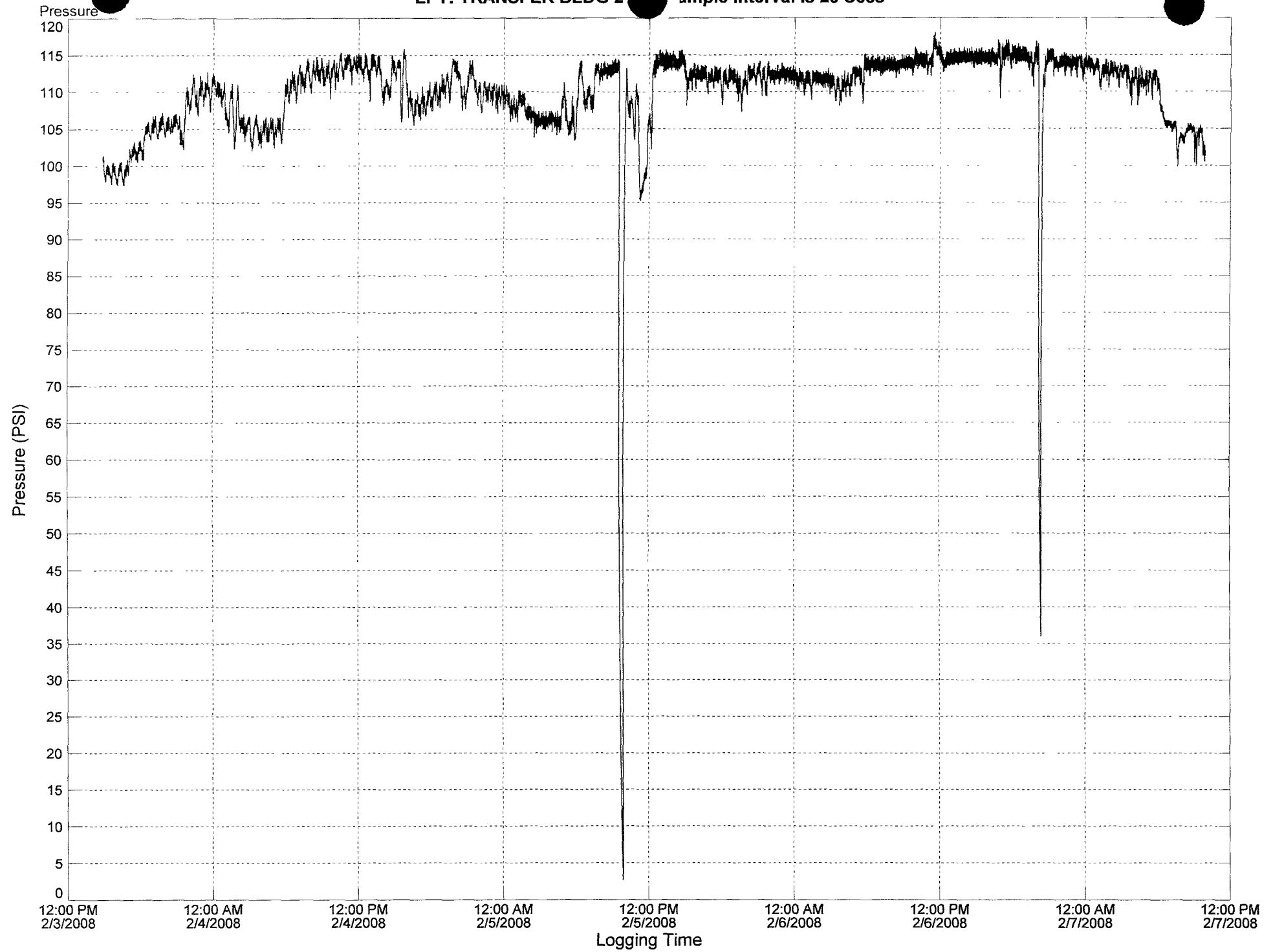
IP12\_006241

LPT: TRANSFER BLDG 4 Sample Interval Is 20 Secs



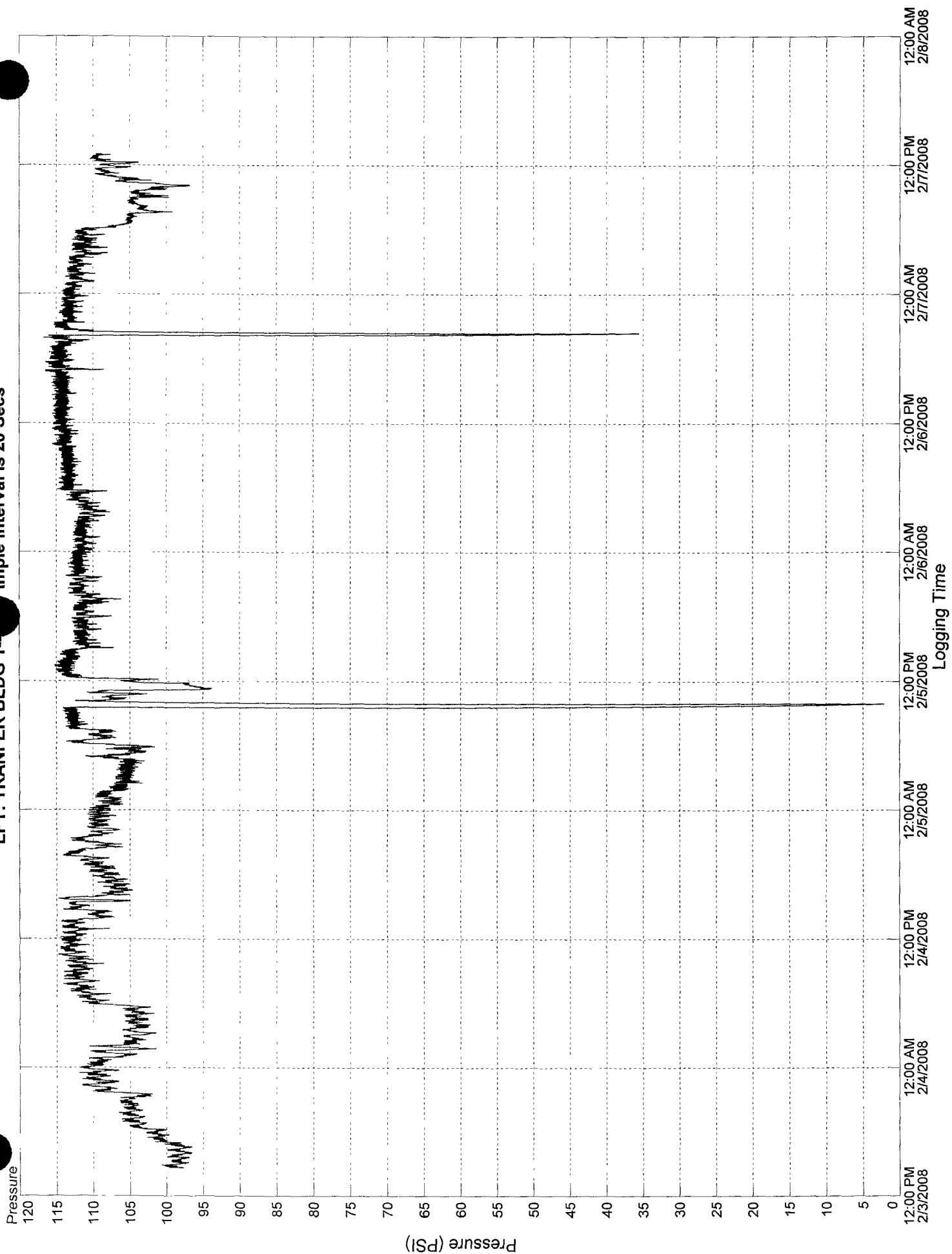
IP12\_006242

LPT: TRANSFER BLDG 2 sample Interval Is 20 Secs



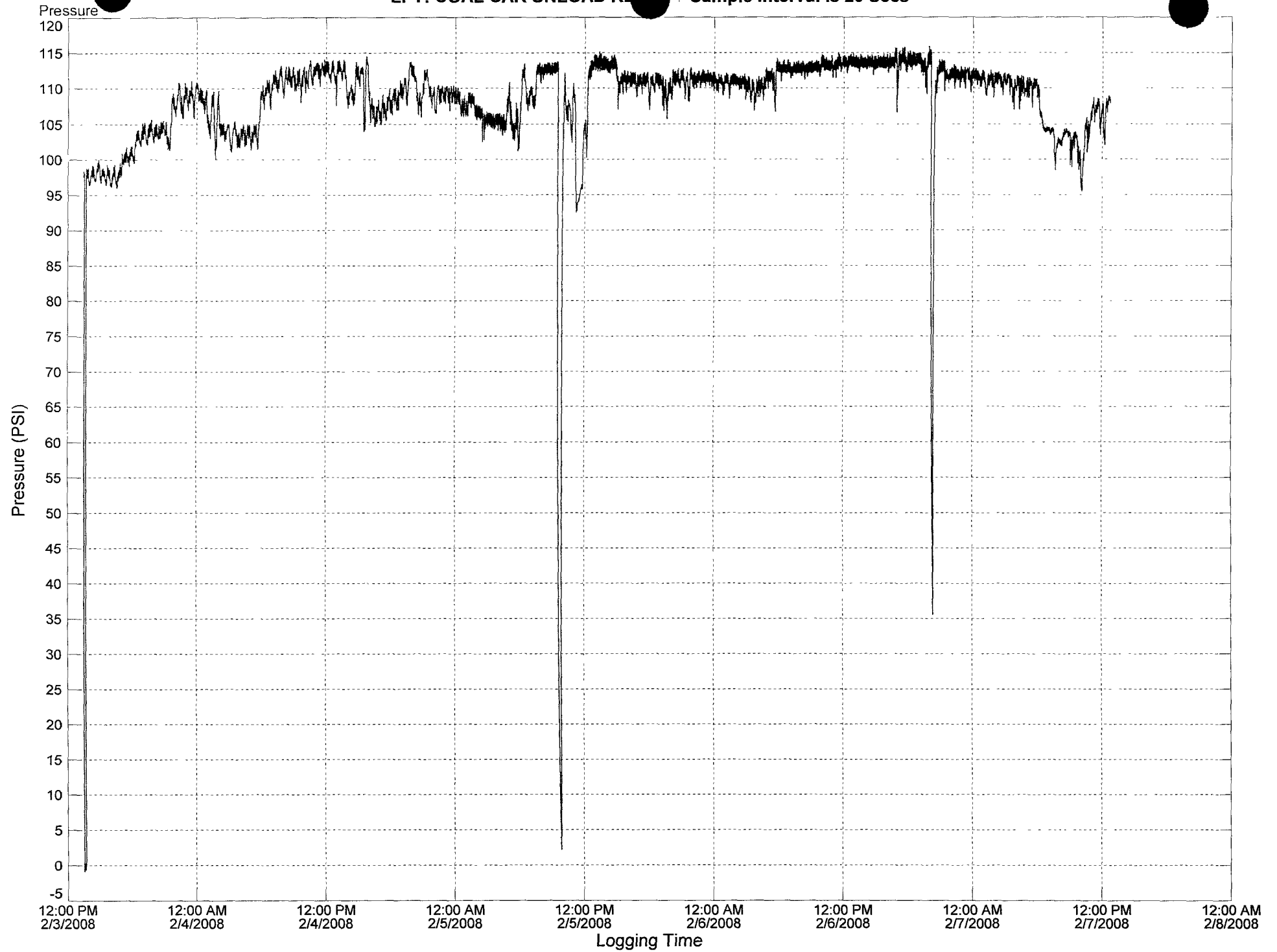
IP12\_006243

LPT: TRANSFER BLDG 1-- Sample Interval Is 20 Secs



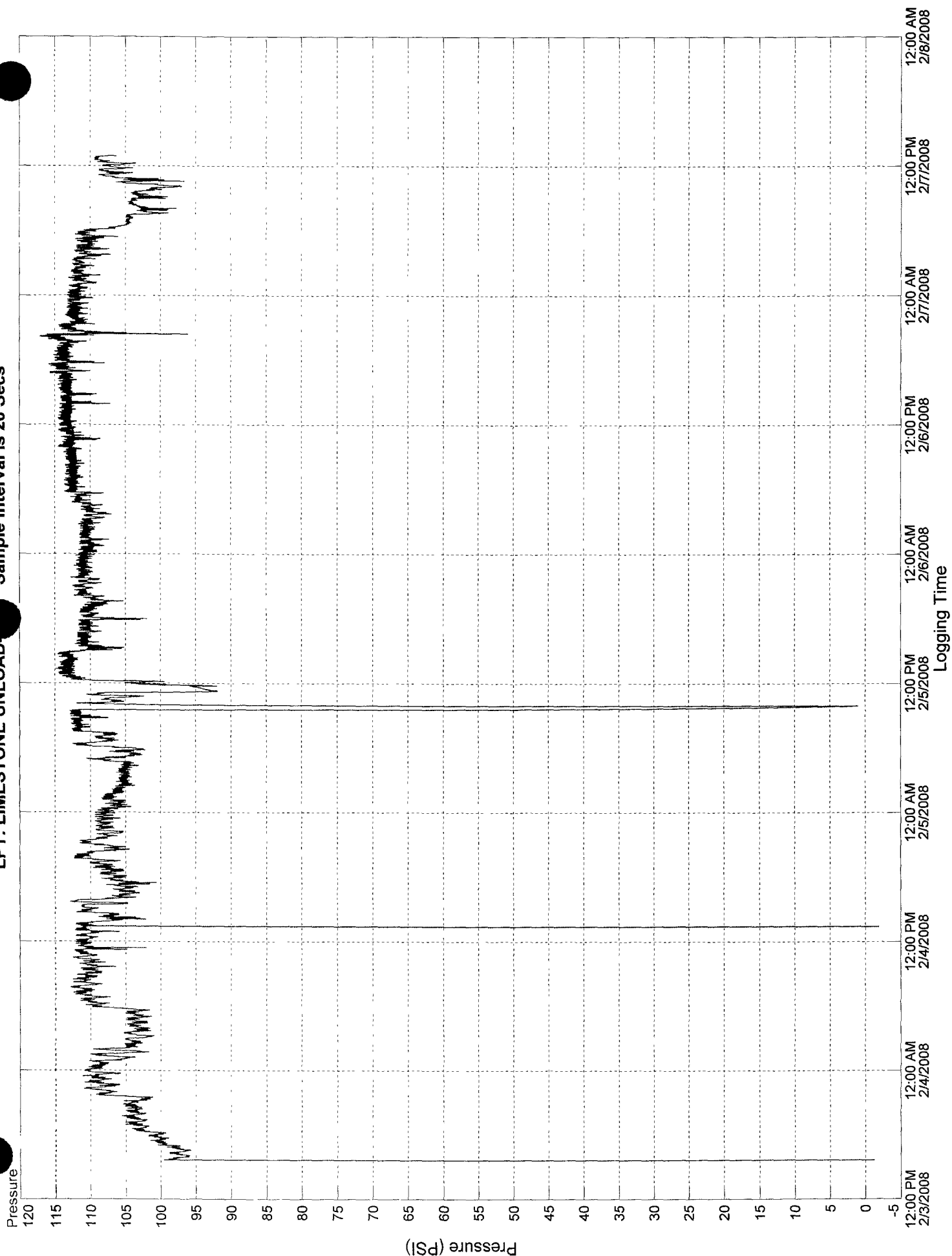
IP12\_006244

LPT: COAL CAR UNLOAD RE Sample Interval Is 20 Secs



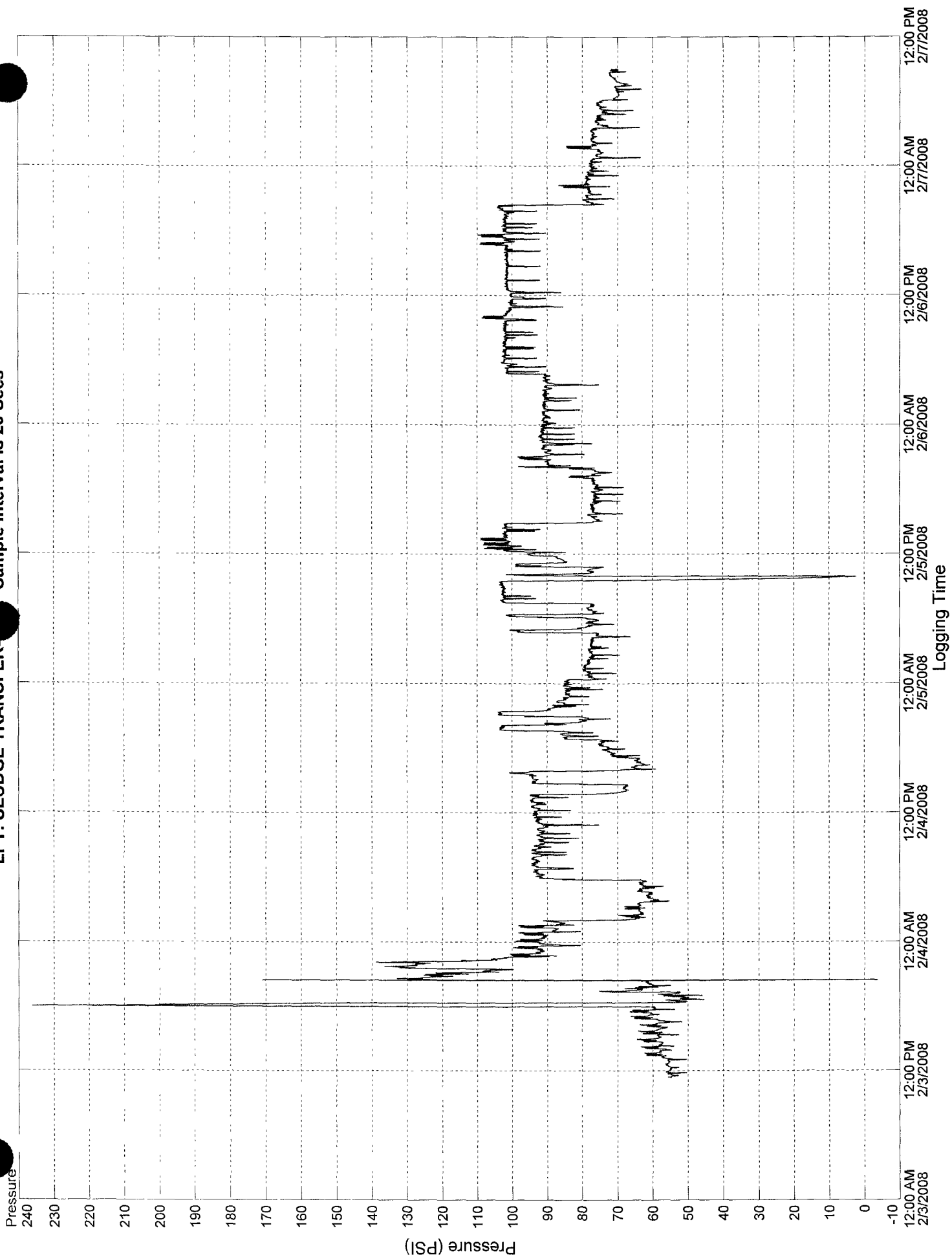
IP12\_006245

LPT: LIMESTONE UNLOAD- Sample Interval Is 20 Secs



IP12\_006246

LPT: SLUDGE TRANSFER- Sample Interval Is 20 Secs



IP12\_006247

LPT: STACK---12. s Interval is 20 Secs

Pressure

120

115

110

105

100

95

90

85

80

75

70

65

60

55

50

45

40

35

30

25

20

15

10

5

0

-5

Pressure (PSI)

12:00 PM  
2/3/2008

12:00 AM  
2/4/2008

12:00 PM  
2/4/2008

12:00 AM  
2/5/2008

12:00 PM  
2/5/2008

12:00 AM  
2/6/2008

12:00 PM  
2/6/2008

12:00 AM  
2/7/2008

12:00 PM  
2/7/2008

12:00 AM  
2/8/2008

Logging Time

IP12\_006248

LPT: WTR TREAT REC--- Sample Interval Is 20 Secs

Pressure

115

110

105

100

95

90

85

80

75

70

65

60

55

50

45

40

35

30

25

20

15

10

5

0

Pressure (PSI)

12:00 PM  
2/3/2008

12:00 AM  
2/4/2008

12:00 PM  
2/4/2008

12:00 AM  
2/5/2008

12:00 PM  
2/5/2008

12:00 AM  
2/6/2008

12:00 PM  
2/6/2008

12:00 AM  
2/7/2008

12:00 PM  
2/7/2008

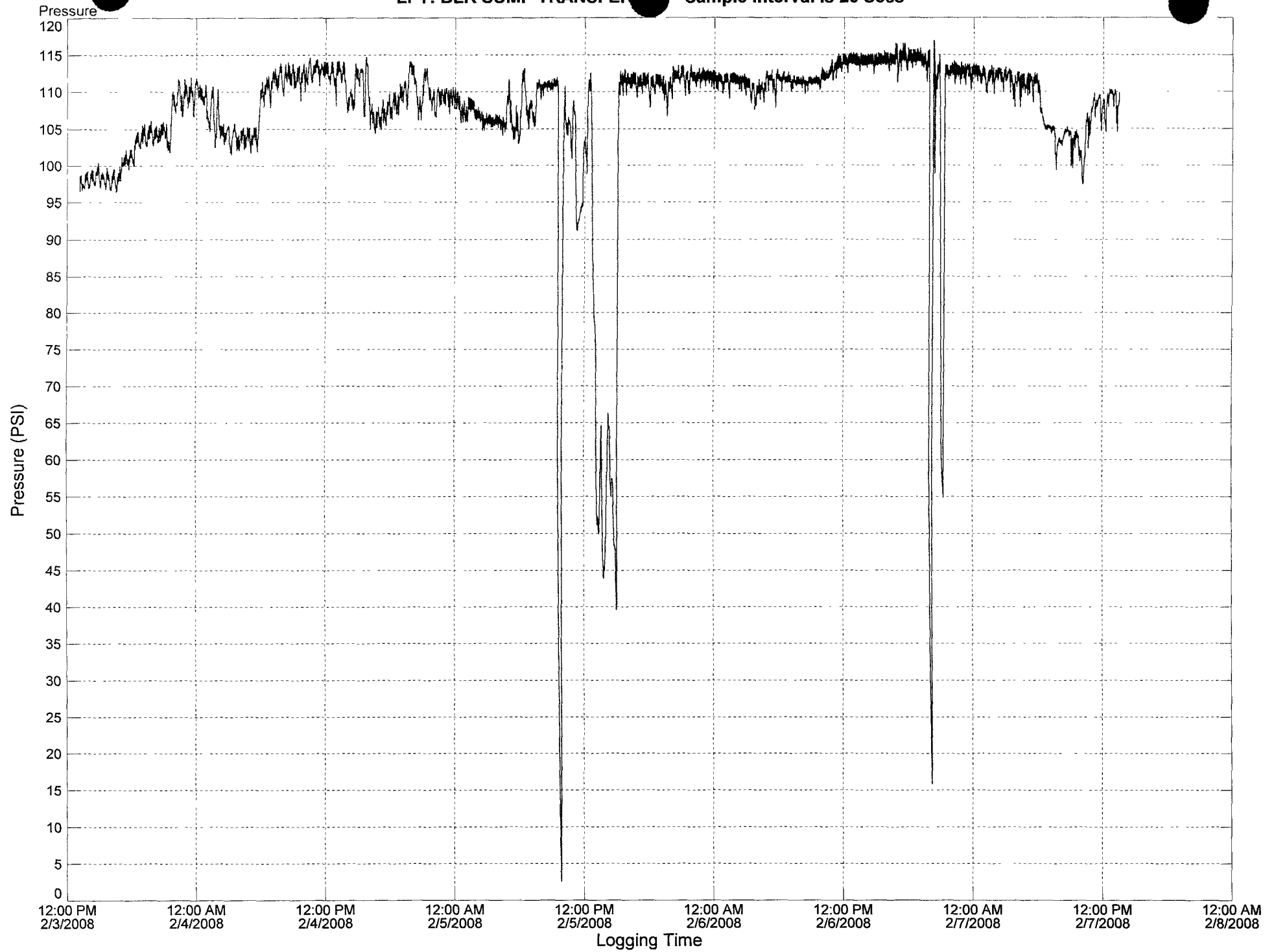
12:00 AM  
2/8/2008

Logging Time

IP12\_006249

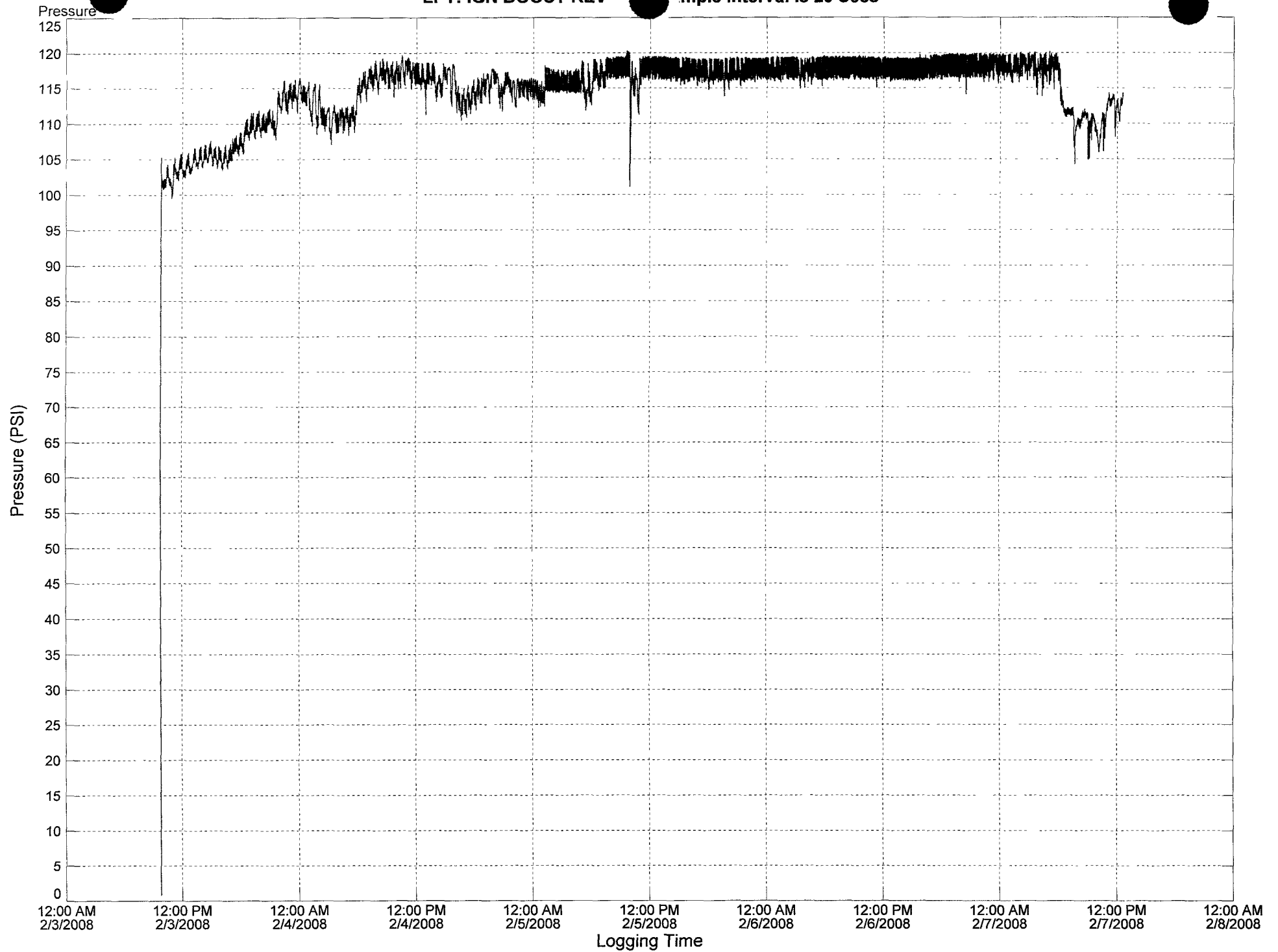
LPT: BLR SUMP TRANSFER

Sample Interval Is 20 Secs



IP12\_006250

LPT: IGN BOOST REV--- Sample Interval Is 20 Secs



IP12\_006251

LPT: LIME PREP REC---1 Sample Interval Is 20 Secs

Pressure

120

115

110

105

100

95

90

85

80

75

70

65

60

55

50

45

40

35

30

25

20

15

10

5

0

Pressure (PSI)

12:00 AM  
2/3/2008

12:00 PM  
2/3/2008

12:00 AM  
2/4/2008

12:00 PM  
2/4/2008

12:00 AM  
2/5/2008

12:00 PM  
2/5/2008

12:00 AM  
2/6/2008

12:00 PM  
2/6/2008

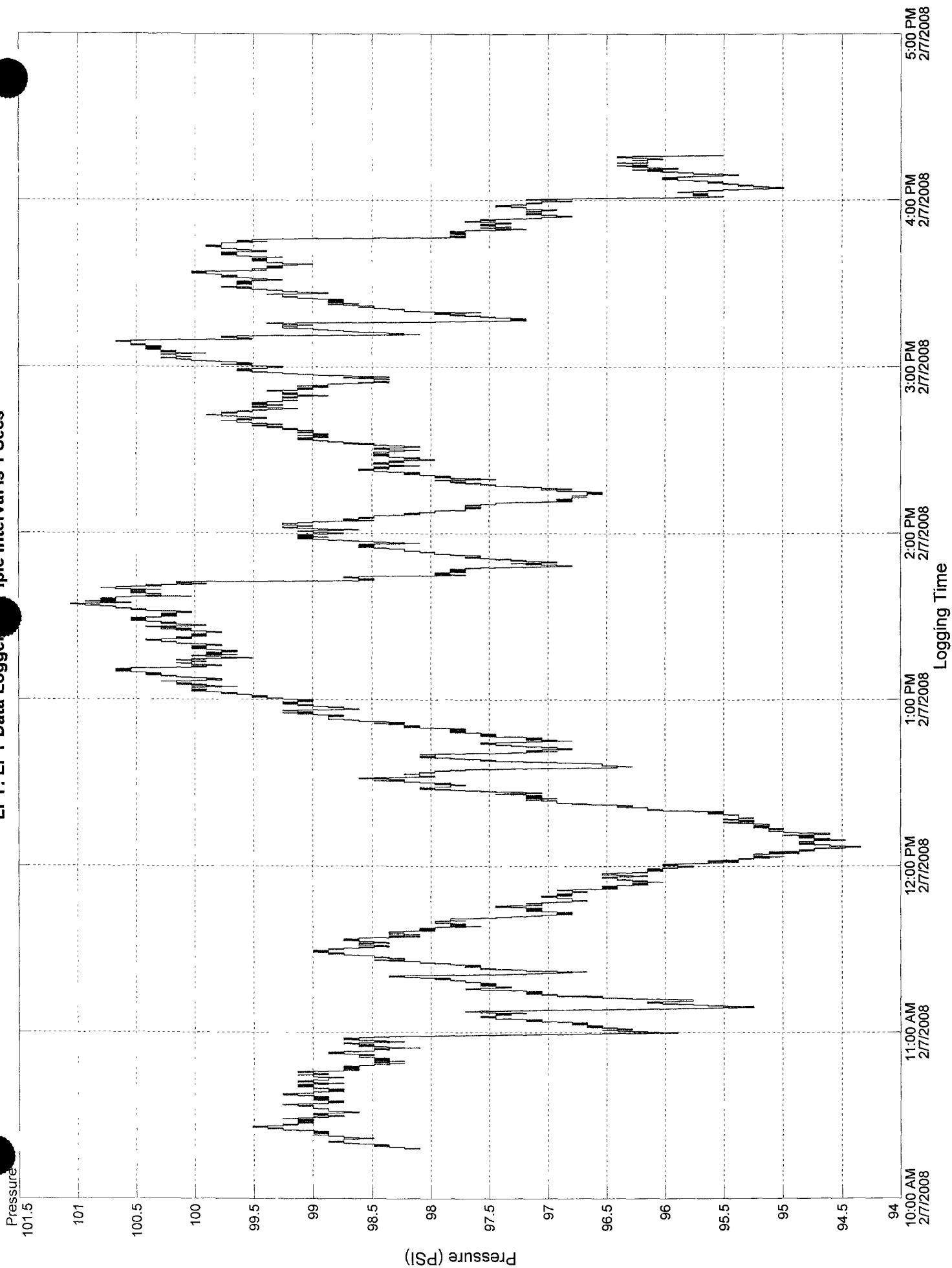
12:00 AM  
2/7/2008

12:00 PM  
2/7/2008

Logging Time

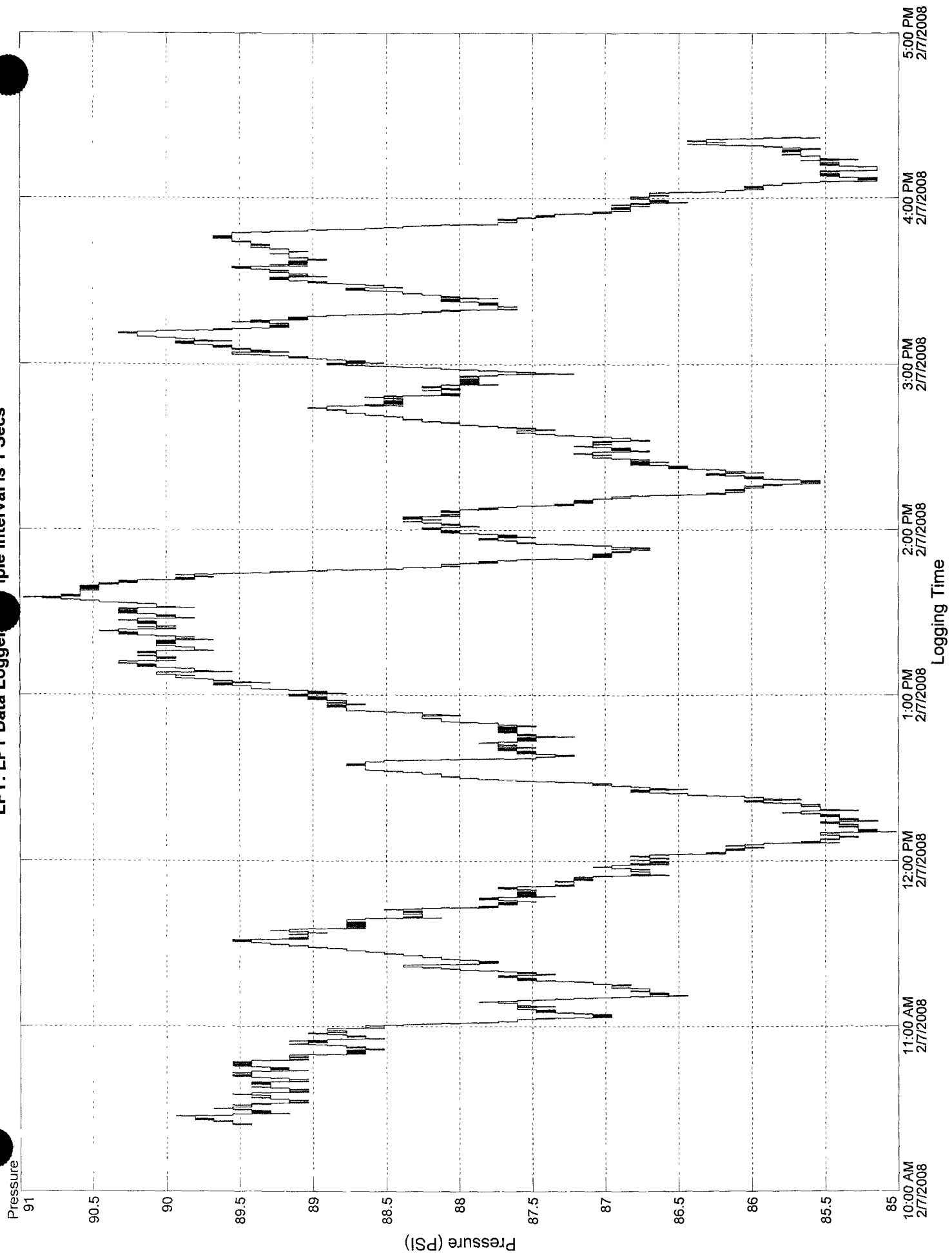
IP12\_006252

LPT: LPT Data Logger Sample Interval Is 1 Secs



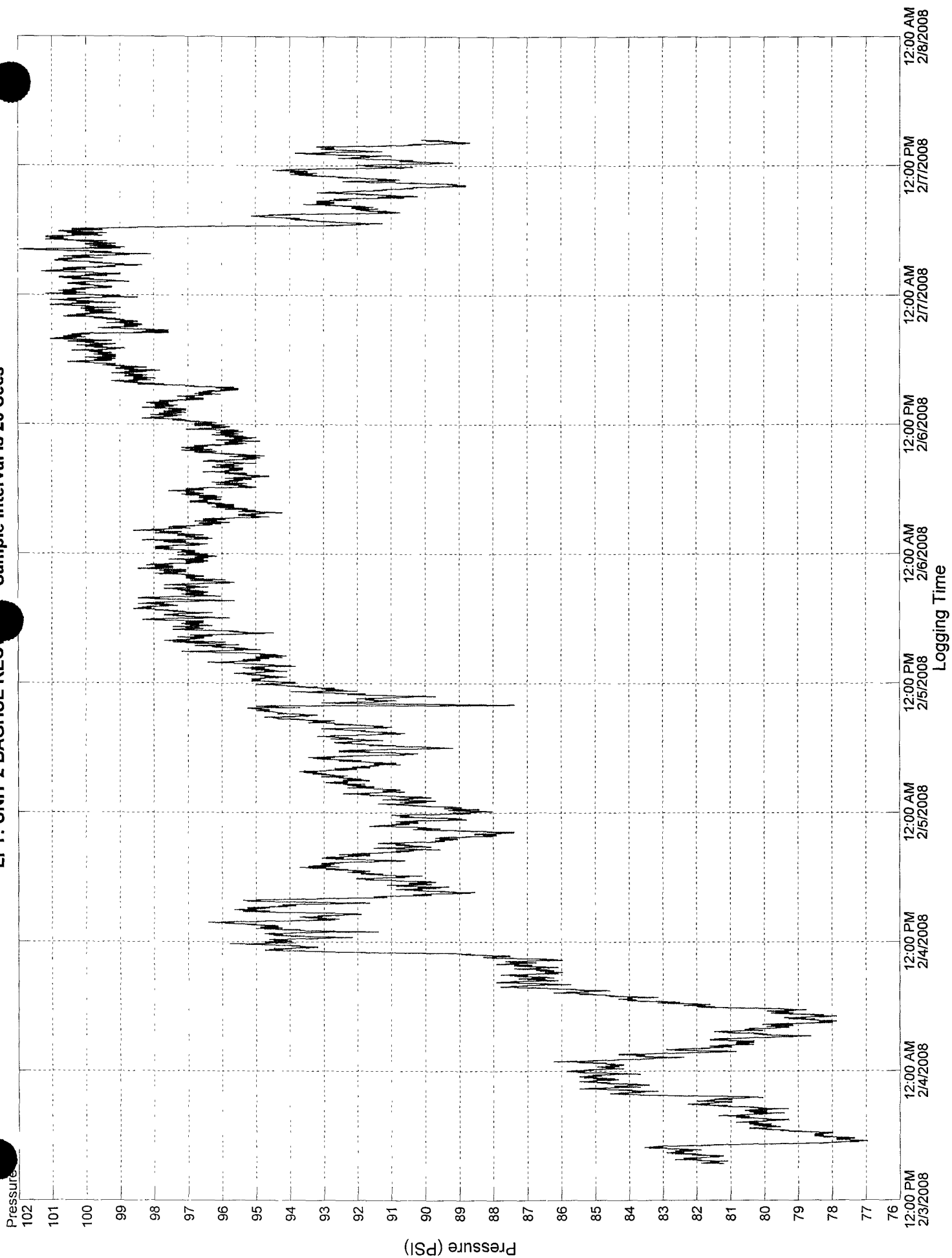
IP12\_006253

LPT: LPT Data Logger Sample Interval Is 1 Secs

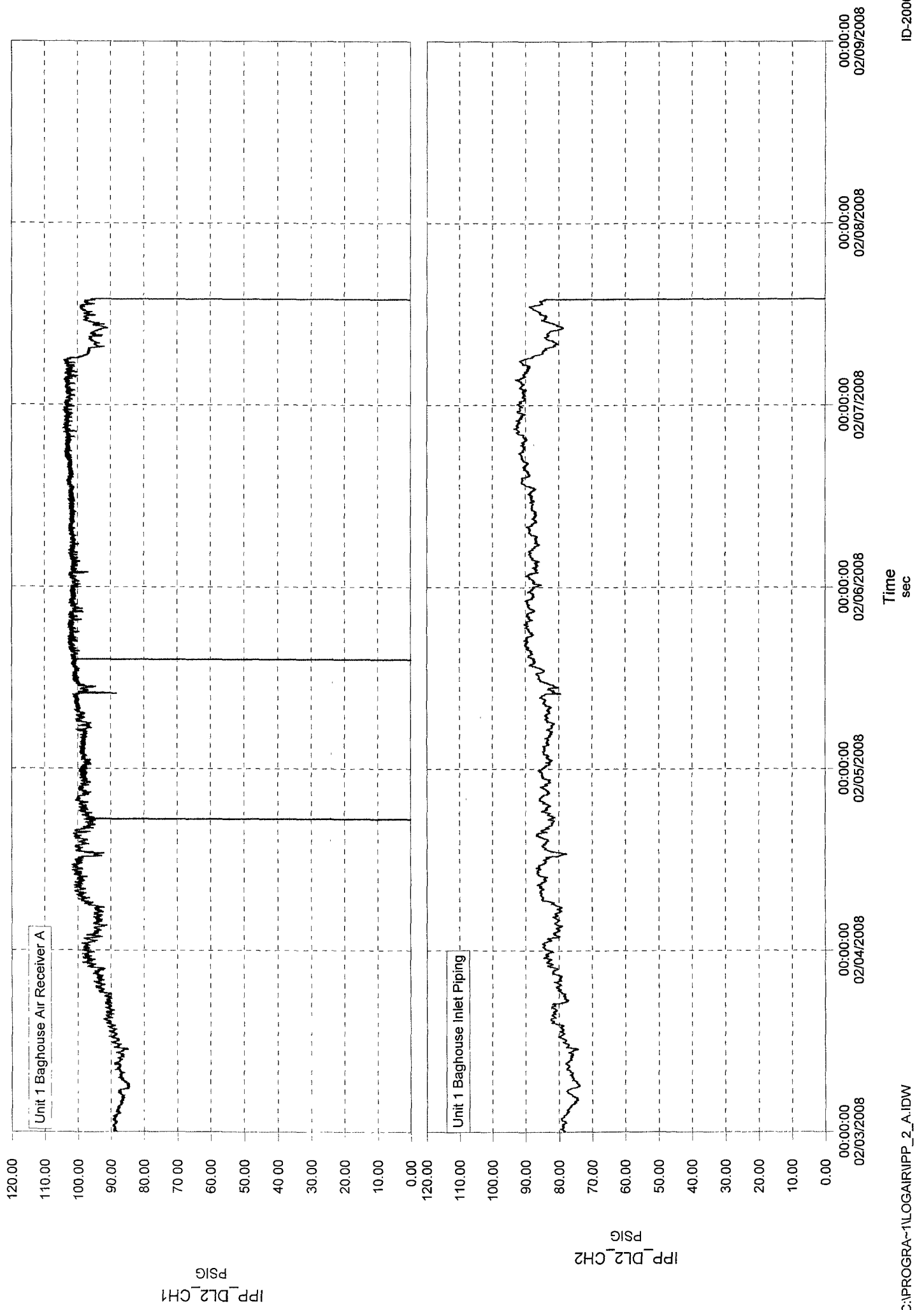


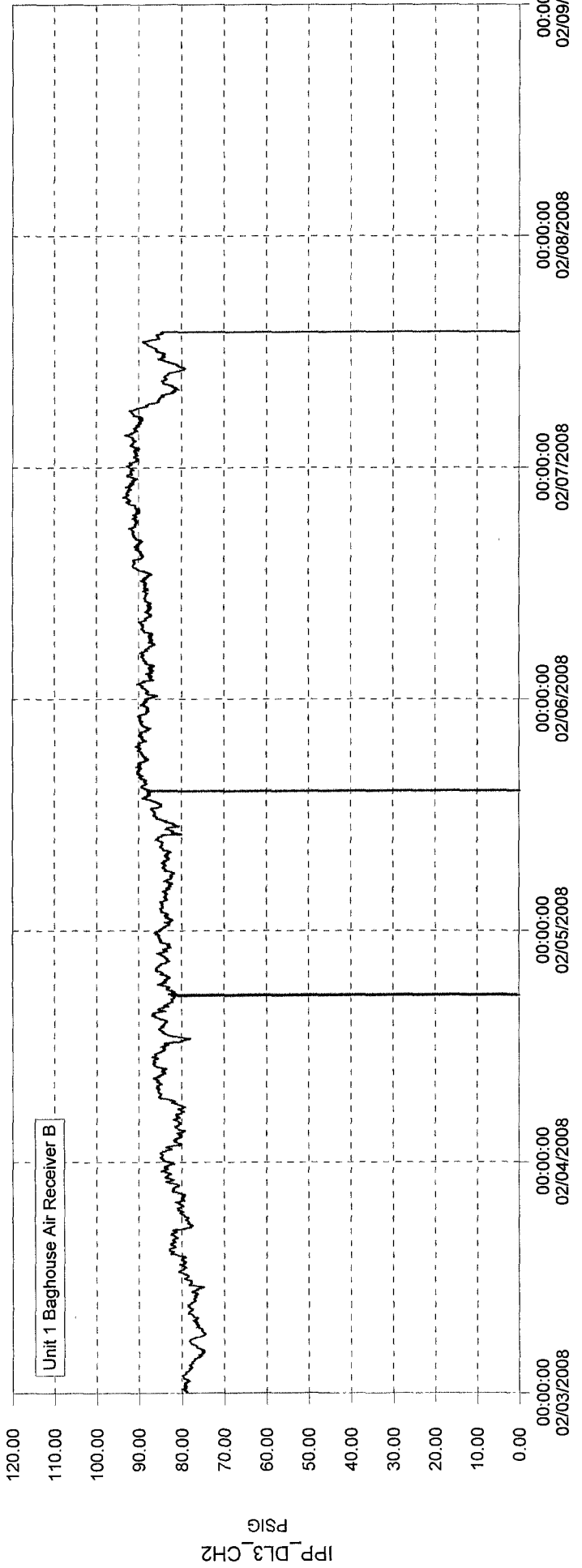
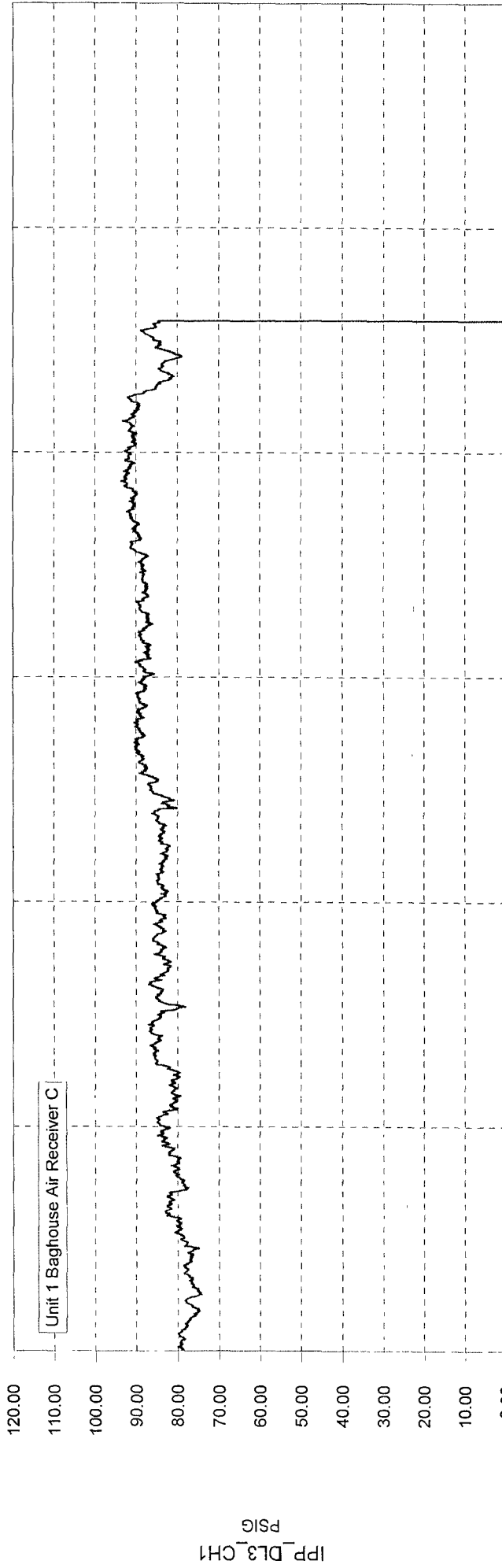
IP12\_006254

LPT: UNIT 2 BAGHSE RECV Sample Interval Is 20 Secs



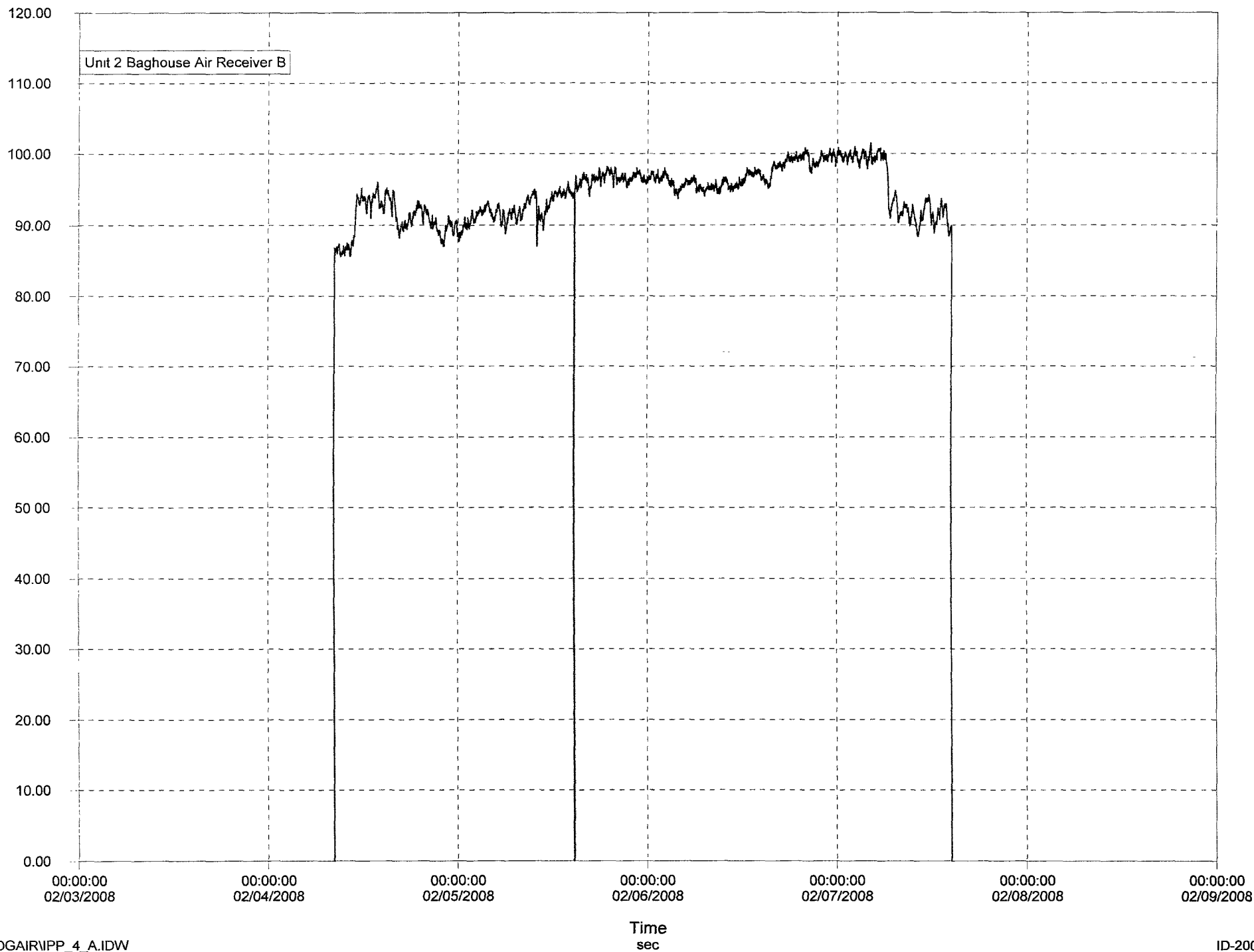
IP12\_006255





IP12\_006258

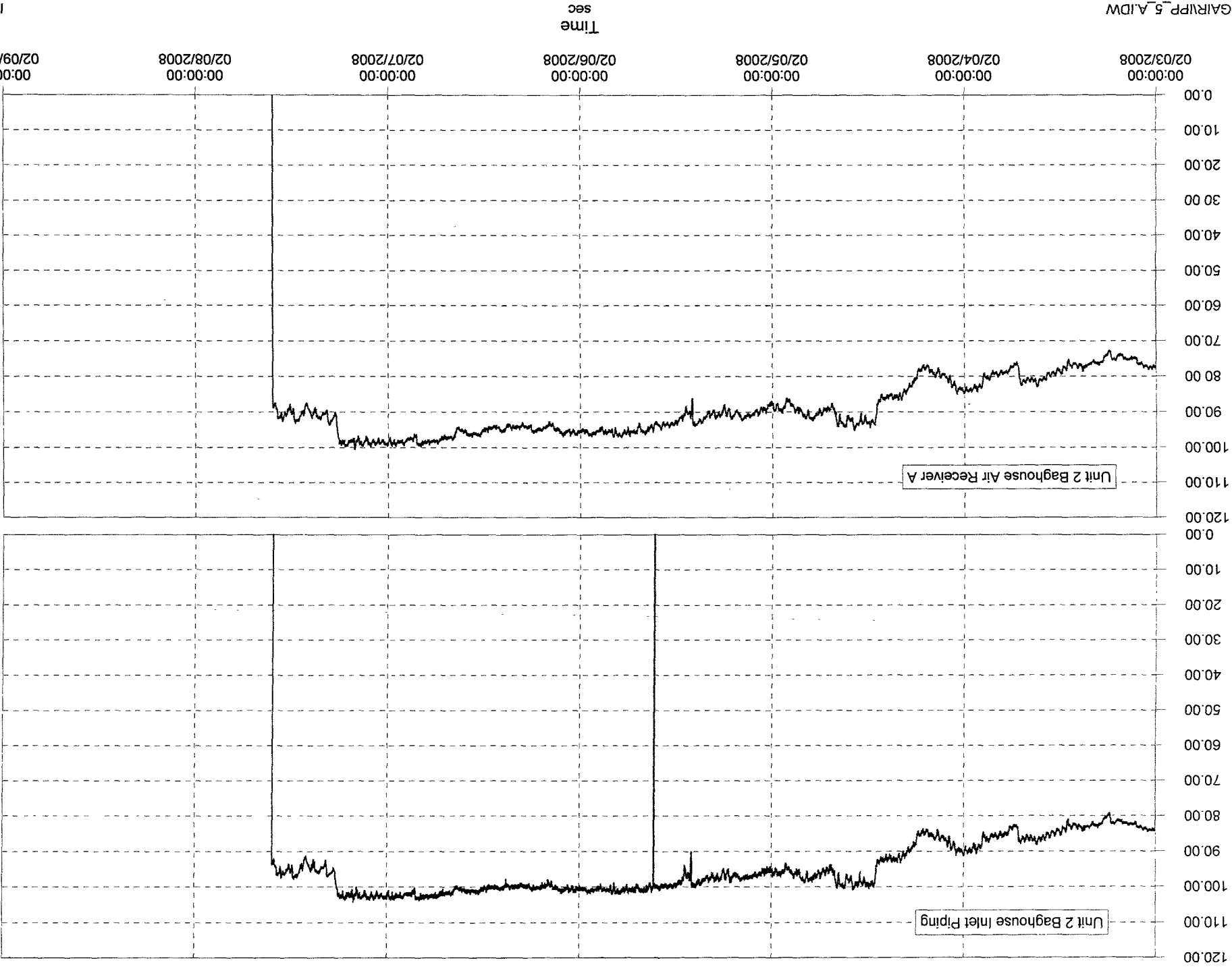
IPP\_DL4\_CH1



IP12\_006259

IPP\_DL5\_CH2  
PSIG

IPP\_DL5\_CH1  
PSIG



The graph displays the pressure of IPP\_4\_CH4 over time. The y-axis represents pressure in psig, ranging from 0.00 to 120.00. The x-axis represents time in seconds, with labels every 30 seconds from 08:20:00 to 08:23:30 on 02/04/2008. The data shows a steady increase in pressure from 85 psig at 08:20:00 to 90 psig at 08:21:00, after which it remains constant at 90 psig until 08:23:30. A text box at 08:21:00 indicates the start of data logging for Unit 2 Baghouse Air Receiver B.

| Time (sec)          | Pressure (psig) |
|---------------------|-----------------|
| 08:20:00 02/04/2008 | 85.00           |
| 08:20:30 02/04/2008 | 87.50           |
| 08:21:00 02/04/2008 | 90.00           |
| 08:21:30 02/04/2008 | 90.00           |
| 08:22:00 02/04/2008 | 90.00           |
| 08:22:30 02/04/2008 | 90.00           |
| 08:23:00 02/04/2008 | 90.00           |
| 08:23:30 02/04/2008 | 90.00           |

MOTOR AIR COMPRESSOR 1A, 1B, 1C + 1D

SPEC 9255  
PROJECT 62.0401

MOTOR DATA TO BE SUBMITTED

MANUFACTURER Westinghouse Electric Corp MODEL 82F52938  
HP 700 VOLTS 6600 PHASE 3 HERTZ 60  
SERVICE FACTOR 1.00 NEMA DESIGN LETTER B FULL LOAD SPEED 3572 RPM

ENCLOSURE: TYPE ODPG FRAME SIZE 581A H

INSULATION SYSTEM: CLASS B STANDARD SEAL X AMB TEMP 40 C

TEMP. RISE 80 C BY RESISTANCE AT SERVICE FACTOR OF 1.0 1.15 Thermalastic Epoxy

FULL LOAD CURRENT 53 AMPS, LOCKED-ROTOR CURRENT 384 AMPS

SPACE HEATER (IF FURNISHED): NUMBER OF UNITS 2 UNIT RATING, WATTS 201

VOLTS 110 PHASE 1

BEARINGS: TYPE Split Sleeve AFBMA L-10 RATING LIFE, NOT LESS THAN NA HRS

LUBRICATION: TYPE Oil SYSTEM Ring Lube

SOUND LEVELS:

SOUND POWER LEVEL  
RE 10<sup>-12</sup> WATTS        dBA  
FREE FIELD

OVERALL MEAN NO-LOAD SOUND PRESSURE LEVEL  
RE 20 MICROPASCALS (0.0002 MICROBARI) REFERENCE DISTANCE  
OF 1 METER 80 2 METERS        :        dBA FREE FIELD

TOTAL MOTOR WT 4850 Approx LBS DESIGN ALTITUDE 4676 FT

FOR MULTISPEED MOTORS:

VARIABLE TORQUE        CONSTANT TORQUE        CONSTANT HORSEPOWER         
MOTOR TERMINAL CONNECTION DIAGRAM NO.        (ATTACH COPY OF DIAGRAM)

FOR WOUND ROTOR MOTORS:

SEC. VOLTS        SEC. AMPS        SEC. RES., OHMS H-M AT 25 C       

FOR MOTORS IN HAZARDOUS LOCATIONS:

MOTOR ENCLOSURE SURFACE TEMPERATURE,        C AT SERVICE FACTOR OF 1.0 1.15  
WILL MOTOR CONTAIN A SURFACE TEMPERATURE CONTROL THERMOSTAT REQUIRING CONNECTION INTO THE MOTOR  
STARTER CONTROL CIRCUIT: YES        NO       

FOR DUST IGNITION-PROOF MOTORS: MOTOR ENCLOSURE SURFACE TEMPERATURE RISE UNDER ANY ABNORMAL  
OPERATING CONDITION INCLUDING OVERLOAD, SINGLE-PHASING, ETC., ASSUMING ENCLOSURE SURFACE  
TEMPERATURE OF 120 C WHEN ABNORMAL CONDITION DEVELOPS:

MINIMUM TIME TO REACH 165 C        SECS

MAXIMUM RATE OF RISE        C IN 5 SECS

ADDITIONAL MOTOR DATA FOR MOTORS LARGER THAN 200 HP AND FOR ALL MOTORS RATED ABOVE 600 VOLTS  
SHALL BE SUBMITTED ON SHEETS 2 AND 3.

BLACK & VEATCH  
CONSULTING ENGINEERS



MOTOR  
INFORMATION SHEET

1C-16

MOTOR Air Compressor 1A, 1B, 1C & 1DSPEC 9255  
PROJECT 62.0401**ADDITIONAL MOTOR DATA TO BE SUBMITTED FOR MOTORS LARGER THAN 200 HP AND FOR ALL MOTORS RATED ABOVE 600 VOLTS**EFFICIENCY, PER CENT ~~GUARANTEED~~ Calculated  
LOAD: 1/2 92.2 3/4 94.7 1/4 96.5 = .943 ESTIMATED  
POWER FACTOR, PER CENT ~~GUARANTEED~~ Calculated  
LOAD: 1/2 85.1 3/4 87.9 1/4 91.4 = .90 ESTIMATEDMINIMUM STARTING VOLTAGE IN  
PER CENT OF RATED VOLTAGE:CALCULATED: 85  
SPECIFIED: 85

OCTAVE BAND MEAN NO-LOAD SOUND PRESSURE LEVEL

RE 20 MICROPASCALS (0.0002 MICROBAR) REFERENCE

DISTANCE OF 1 METER 80 : 2 METERS \_\_\_\_\_OUTLINE DWG. NO. 2D01215

ACCELERATING TIME:

AT RATED VOLTAGE 4.6 SEC.AT MINIMUM SPECIFIED  
STARTING VOLTAGE 7.3 SEC.LOCKED-ROTOR PF 22.1 PER CENTLOCKED-ROTOR TORQUE 968 LB-FTPULL-UP TORQUE 776 LB-FTBREAKDOWN TORQUE 3156 LB-FTTOTAL ROTOR WT 986 APPROX LBSROTOR INERTIA 144 LB-FT<sup>2</sup>OPEN CIR. TIME CONSTANT 2.07 SEC.FOR VERTICAL MOTORS:  
REED FREQUENCY NA HZ

CENTER FREQUENCY, HZ

LEVEL, dBC

31.5

85

63

85

125

85

250

75

500

77

1000

75

2000

73

4000

73

8000

73THERMAL LIMIT CURVE UNDER LOCKED-ROTOR,  
ACCELERATION, AND RUNNING OVERLOAD CONDITIONS;  
AND TIME-CURRENT CURVES DURING ACCELERATION AT  
RATED VOLTAGE AND AT MINIMUM SPECIFIED STARTING  
VOLTAGE CURVE NO. \_\_\_\_\_ (ATTACH CURVE)

LOCKED-ROTOR SAFE STALLED TIME, SECS

RATED VOLTAGE

MINIMUM SPECIFIED  
STARTING VOLTAGEMOTOR INITIALLY AT MAXIMUM SPECIFIED  
AMBIENT TEMPERATURE710.8MOTOR INITIALLY AT SERVICE FACTOR  
LOAD OPERATING TEMPERATURE710.8BLACK & VEATCH  
CONSULTING ENGINEERSMOTOR  
INFORMATION SHEET

1C-17

P-EL-154F

A 10/05/83 RGL/ma

IP12\_006262

MOTOR AIR COMPRESSOR 1A, 1B, 1C + 1DSPEC 9255  
PROJECT 62.0401**ADDITIONAL MOTOR DATA TO BE SUBMITTED FOR MOTORS LARGER THAN 200 HP AND FOR ALL MOTORS RATED ABOVE 600 VOLTS.**SPEED-TORQUE-CURRENT CURVES AT RATED VOLTAGE  
AND AT MINIMUM SPECIFIED STARTING VOLTAGE, IN  
PER CENT OF FULL LOAD OR CURVE NUMBERS \_\_\_\_\_  
(ATTACH CURVES)

|  | RATED VOLTAGE   |                 | MINIMUM SPECIFIED<br>STARTING VOLTAGE |                 |
|--|-----------------|-----------------|---------------------------------------|-----------------|
|  | TORQUE<br>LB-FT | CURRENT<br>AMPS | TORQUE<br>LB-FT                       | CURRENT<br>AMPS |
| AT SPEED CORRESPONDING TO MAXIMUM TORQUE | <u>1026</u>     | <u>53</u>       | <u>1026</u>                           | <u>62</u>       |
| AT <del>80</del> SPEED <u>90%</u>        | <u>1275</u>     | <u>325</u>      | <u>1295</u>                           | <u>270</u>      |
| AT 60% SPEED                             | <u>1005</u>     | <u>355</u>      | <u>691</u>                            | <u>292</u>      |
| AT LOCKED ROTOR                          | <u>968</u>      | <u>324</u>      | <u>664</u>                            | <u>317</u>      |

NUMBER OF SUCCESSIVE STARTS

MOTOR INITIALLY AT MAXIMUM SPECIFIED  
AMBIENT TEMPERATURE (COLD)  
MOTOR INITIALLY AT SERVICE FACTOR LOAD  
OPERATING TEMPERATURE (HOT)

AT RATED VOLTAGE

2  
1AT MINIMUM SPECIFIED  
STARTING VOLTAGE2  
1COOLING PERIOD REQUIRED AFTER COMPLETION OF  
THE PRECEDING MAXIMUM NUMBER OF SUCCESSIVE  
STARTS BEFORE MAKING ADDITIONAL STARTS BASED  
ON THE FOLLOWING COOLING CONDITIONS, MINUTESMOTOR RUNNING AT SERVICE FACTOR LOAD  
MOTOR RUNNING WITH DRIVEN EQUIPMENT  
UNLOADED  
MOTOR DE-ENERGIZED, COASTED TO STOP,  
AND LEFT IDLE

|                |                |
|----------------|----------------|
| <u>15 min.</u> | <u>15 min.</u> |
| <u>15 min.</u> | <u>15 min.</u> |
| <u>45 min.</u> | <u>45 min.</u> |

FOR MOTORS FOR WHICH DIFFERENTIAL PROTECTION IS SPECIFIED, STATOR CONN: VYE, DELTA

FOR MOTORS REQUIRING INTEGRAL WATER COOLING OF BEARING OIL:

FLOW OF COOLING WATER REQUIRED \_\_\_\_\_ GPM  
MAXIMUM COOLING WATER INLET TEMPERATURE \_\_\_\_\_ C

FOR MOTORS CONNECTED TO EXTERNAL LUBRICANT RECIRCULATING SYSTEM:

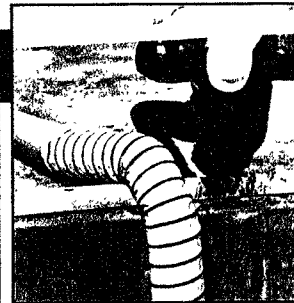
FLOW OF OIL REQUIRED TO MOTOR BEARINGS: \_\_\_\_\_ GPM AT \_\_\_\_\_ C MAXIMUM AT \_\_\_\_\_ PSI  
BEARING HEAT REJECTION TO OIL \_\_\_\_\_ BTU/HR  
OUTLET TEMPERATURE OF OIL FROM MOTOR BEARINGS: \_\_\_\_\_ C MAXIMUMFOR HORIZONTAL MOTORS RATED 1000 HP AND LARGER AT 3600 RPM AND FOR ALL MOTORS RATED 3500 HP AND  
LARGER AT ALL SPEEDS:CRITICAL SPEEDS: FIRST CRITICAL \_\_\_\_\_ RPM  
SECOND CRITICAL \_\_\_\_\_ RPM**BLACK & VEATCH**  
CONSULTING ENGINEERSMOTOR  
INFORMATION SHEET

1C-18

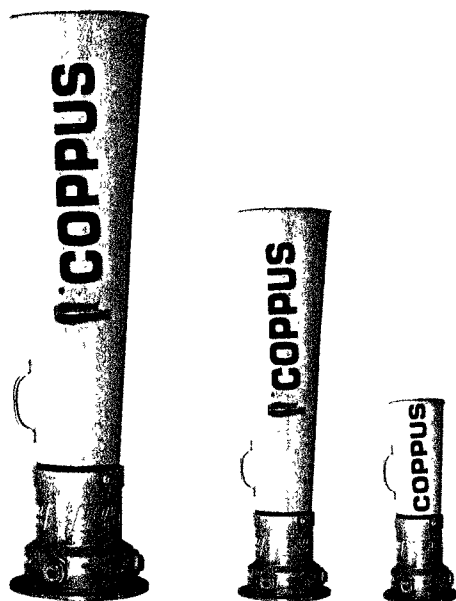
P-EL-139D

A 10/05/83 R. Clancy

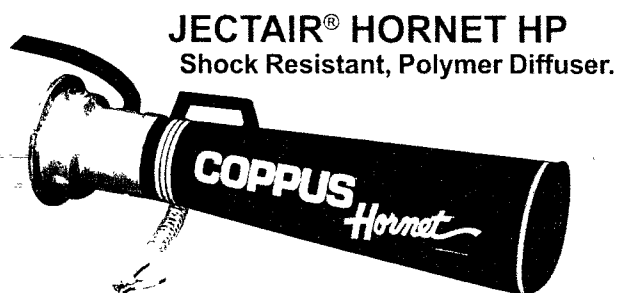
IP12\_006263



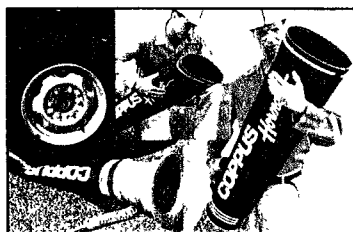
## JECTAIR® HI Performer (HP) and JECTAIR Hornet HP



NOTE: Maximum operating pressure is 150 PSIG



**JECTAIR® HORNET HP**  
Shock Resistant, Polymer Diffuser.



The Jectair Hornet HP features a light weight, shock resistant, conductive polymer diffuser that is virtually indestructible.

- Available in 3 sizes: 3S-HP, 3-HP & 6-HP
- Polymer safely dissipates static electric charges.
- Diffuser is constructed of linear low density polyethylene, rated UL 94-V2 with maximum operating temperature of 160 °F (93 °C)

### High Performance, highly efficient Venturi Air Movers

The unmatched performance of the COPPUS Jectair HPs is recognized throughout the industry. When compared to older style air horns, the patented air mixing chamber of the Jectair HP produces **14% more air while consuming 26% less compressed air.** (see performance/efficiency charts on following page).

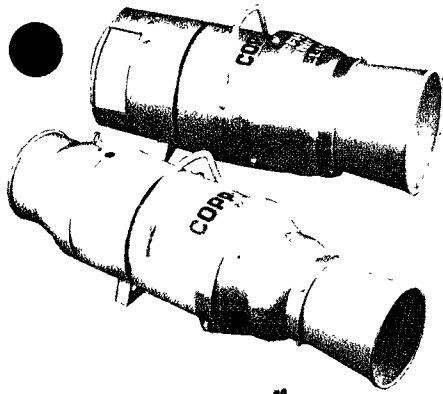
#### FEATURES:

- Available in 5 sizes: 9, 8, 6-HP, 3-HP, 3S-HP
- HI Performer (HP) and Hornet models available in 3 sizes: 6-HP, 3-HP and 3S-HP
- Air flows range from 1370 to 8900 cfm (2,328 to 15,121m³/hr)
- Induction ratios up to 40:1.
- Multiple expansion nozzles machined into housing.
- High static pressure capabilities.
- Diffuser material available in steel, aluminum or shock resistant polymer (Hornet HP).
- Steel diffusers protected by safety yellow, epoxy base powder coating with conductive filler to help dissipate static charge.
- No moving parts - Virtually maintenance free.
- Static bonding cable with spring tension grip and replaceable contact tips standard on all models.
- Accepts flexible duct on diffuser end (optional duct adapter for inlet end available).
- Tripod for stationary mounting available (see Accessories on page 6 for more information).

#### OPERATING PRINCIPLE:

Compressed air or steam is admitted to the Jectair through a single inlet connection in the housing leading to the mixing chamber. The air or steam jetted from the nozzle creates a Venturi action which induces a large volume of surrounding air to enter the Jectair through the inlet bell. The air is then discharged at high velocity through the horn shaped diffuser.

NOTE: Operating efficiency is dependent upon compressed air volume and pressure (see performance/efficiency charts on following page).

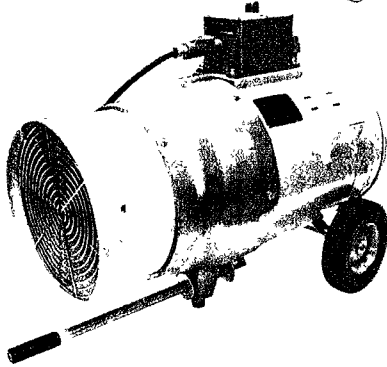


### VANO 175CV and 250CV

Vaneaxial ventilators with the power to move large volumes of air through long runs of flexible duct with minimal loss of air flow. Available with Totally Enclosed, or Explosion Proof electric motor drives. Folding tripod and transport cart accessories available.

**Suggested uses:** Ventilation of tanks, tank cars, manholes, vats, and airplane compartments; welding and toxic fume exhaust; product, equipment & personnel cooling.

**Fan Type:** Vaneaxial. **Drive:** Electric. **Duct:** 8" & 12" (203 & 305 mm).  
**Performance Range:** 1,500 to 3,000 cfm (2,549 to 5,098 m<sup>3</sup>/hr).

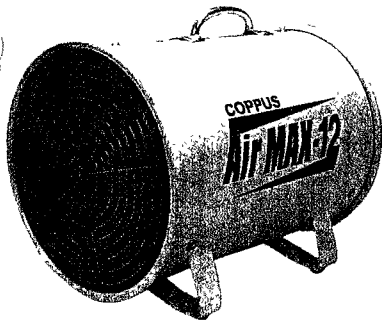


### VANO 500

Provides unsurpassed airflow performance with less than 14% loss in CFM air flow over a 50' flexible duct run. Exceptional, heavy-duty design mounted on 10" wheels with sturdy built-in handle for easy transport to job site. Readily accepts duct on supply or exhaust ends. Totally enclosed or Explosion Proof electric motor drives available.

**Suggested uses:** Ideal for ventilating large confined spaces such as tanks, towers, boilers and underground vaults.

**Fan Type:** Vaneaxial. **Drive:** Electric. **Performance:** 5,000 cfm (8,495 m<sup>3</sup>/hr)  
**Duct:** 16" (406 mm)



### AIR MAX-12

Low profile, high performance Vaneaxial blower. Ideal for fresh air delivery and fume removal in confined areas. Duct able on both ends. Integral motor switch and GFCI protection at plug end. All steel construction with heavy duty 3/4 HP motor.

**Suggested uses:** Welding, painting and coatings, underground and other restricted areas where fume removal and fresh air supply are needed.

**Fan Type:** Vaneaxial **Drive:** Electric. **Duct:** 12" (305mm)  
**Performance:** 2,200 cfm (3,737 m<sup>3</sup>/hr)



### CADET

#### Utility and Light Industrial Air Movers

Ideal for underground utility maintenance and light industrial air supply and exhaust applications. Rugged, lightweight polymer housing is virtually indestructible. Totally Enclosed or Explosion Proof models available.

Available in Vaneaxial and centrifugal designs. Flexible duct and other accessories available. Request the Cadet Catalog for complete product information.

**Suggested uses:** Air supply or fume exhaust applications for underground utility work or light manufacturing applications.

**Fan Type:** Vaneaxial & Centrifugal. **Drive:** Electric, Air and Gasoline  
**Performance Range:** 800 to 1,300 cfm (1,360 to 2,209 m<sup>3</sup>/hr) **Duct:** 8"

**Please note:** Performance listings are maximum CFM Free-Air ratings.

## EFFICIENCY PERFORMANCE

$$\text{INDUCTION RATIO} = \frac{\text{CFM of Total AirFlow}}{\text{CFM of Compressed Air}}$$

| Inlet Pressure       | Model | Air Flow<br>cfm (m³/hr) | Air Consumed<br>(scfm) | Induction Ratio |
|----------------------|-------|-------------------------|------------------------|-----------------|
| 60 psig<br>4,2kg/cm² | 3S-HP | 1370<br>2328            | 47<br>80               | 29.1            |
|                      | 3-HP  | 1520<br>2595            | 47<br>80               | 32.3            |
|                      | 6-HP  | 3980<br>6762            | 98<br>167              | 40.6            |
|                      | 8     | 5600<br>9515            | 178<br>302             | 31.5            |
|                      | 9     | 6880<br>11,096          | 265<br>450             | 25.8            |
| 80 psig<br>5,6kg/cm² | 3S-HP | 1530<br>2600            | 61<br>104              | 25.1            |
|                      | 3-HP  | 1700<br>2888            | 61<br>104              | 27.8            |
|                      | 6-HP  | 4500<br>7645            | 126<br>214             | 35.7            |
|                      | 8     | 6250<br>10,620          | 233<br>396             | 26.8            |
|                      | 9     | 8000<br>13,592          | 336<br>571             | 23.8            |
| 100 psig<br>7kg/cm²  | 3S-HP | 1660<br>2820            | 72<br>122              | 25.8            |
|                      | 3-HP  | 1860<br>3160            | 72<br>122              | 23.0            |
|                      | 6-HP  | 4870<br>8274            | 153<br>260             | 31.8            |
|                      | 8     | 6750<br>11,469          | 282<br>479             | 23.9            |
|                      | 9     | 8900<br>15,121          | 410<br>697             | 21.7            |

Maximum operating pressure of 150 psig on compressed air or steam.

## BLOCKED TIGHT STATIC PRESSURE

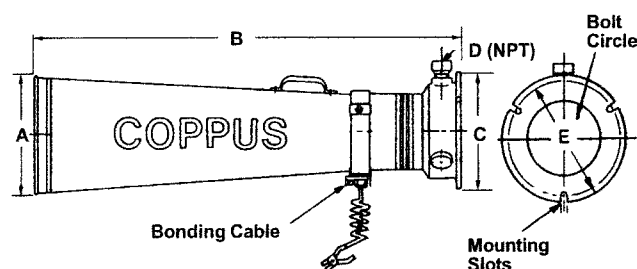
| Model | Inlet Pressure       |                      |                     |
|-------|----------------------|----------------------|---------------------|
|       | 60 psig<br>4,2kg/cm² | 80 psig<br>5,6kg/cm² | 100 psig<br>7kg/cm² |
| 3S-HP | 5.8"<br>147mm        | 7.5"<br>191mm        | 8.9"<br>224mm       |
| 3-HP  | 5.8"<br>147mm        | 7.5"<br>191mm        | 8.9"<br>224mm       |
| 6-HP  | 4.3"<br>109mm        | 5.6"<br>132mm        | 6.7"<br>170mm       |
| 8     | 3.9"<br>99mm         | 5.2"<br>132mm        | 6.2"<br>157mm       |
| 9     | 5.5"<br>140mm        | 6.8"<br>173mm        | 7.7"<br>196mm       |

## PERFORMANCE THROUGH VARIOUS LENGTHS OF FLEXIBLE DUCT AT 80 psig (7kg/cm²)

High static pressure capabilities of the Jectair HP permits use of long runs of flexible duct on outlet or inlet diffuser.

| Model | Duct Diameter | Free Air<br>cfm/m³/hr | 20'/6m<br>cfm/m³/hr | 30'/9m<br>cfm/m³/hr | 40'/12m<br>cfm/m³/hr | 50'/15m<br>cfm/m³/hr |
|-------|---------------|-----------------------|---------------------|---------------------|----------------------|----------------------|
| 3-HP  | 8"<br>203mm   | 1700<br>2888          | 1550<br>2634        | 1480<br>2515        | 1410<br>2396         | 1350<br>2294         |
| 6-HP  | 12"<br>305mm  | 4500<br>7645          | 4020<br>6830        | 3860<br>6558        | 3715<br>6312         | 3580<br>6083         |
| 8     | 14"<br>356mm  | 6250<br>10,620        | 5550<br>9431        | 5280<br>8972        | 5050<br>8581         | 4850<br>8241         |
| 9     | 14"<br>356mm  | 8000<br>13,592        | 6850<br>11,640      | 6550<br>11,130      | 6250<br>10,620       | 6000<br>10,195       |

## DIMENSIONS



| Model | A           | B            | C           | D         | Mounting Slots |     |           | Wt.             |
|-------|-------------|--------------|-------------|-----------|----------------|-----|-----------|-----------------|
|       |             |              |             |           | E              | No. | Width     |                 |
| 3S-HP | 6.0<br>152  | 16.5<br>419  | 7.5<br>190  | 1/2<br>13 | 6.5<br>165     | 3   | 0.4<br>10 | 6lbs<br>2,7kg   |
| 3-HP  | 7.3<br>185  | 33.0<br>838  | 7.5<br>190  | 1/2<br>13 | 6.5<br>165     | 3   | 0.4<br>10 | 9lbs<br>4,1kg   |
| 6-HP  | 12.0<br>305 | 44.2<br>1123 | 11.5<br>292 | 1<br>25   | 10.8<br>274    | 3   | 0.4<br>10 | 21lbs<br>9,5kg  |
| 8     | 14.0<br>356 | 46.0<br>1168 | 14.3<br>363 | 1<br>25   | 13.5<br>343    | 3   | 0.5<br>13 | 35lbs<br>15,9kg |
| 9     | 14.0<br>356 | 46.0<br>1168 | 16.8<br>427 | 1<br>25   | 15.3<br>387    | 10  | 0.9<br>23 | 42lbs<br>19,0kg |

## SAFETY PRECAUTIONS

- Use Bonding Cables (standard on all COPPUS Jectairs) when operating in hazardous locations to prevent static electrical discharges.
- Secure unit before admitting compressed air (or steam) to prevent damage or injury from high reaction force.
- Do not allow solid objects or loose debris to enter inlet housing during operation.
- When exhausting fumes from enclosed vessels, take care not to create vacuum conditions that could collapse vessel.

SHAP DATA CORRECTED TO JOB SITE COND.  
 B211523B-2 TEST DATE 7/19/83 N-3570-APP  
 PB = 12.2 PSIA P<sub>IN</sub> = 12.0 PSIA R.H. 20%  
 T<sub>IN</sub> = 100°F T<sub>W</sub> = 105°F  
 DEPT. OF WATER & POWER OF THE CITY OF L.A.  
 FOR: INTERMOUNTAIN POWER AGENCY

NOMINAL PERFORMANCE  
 3100 ICFM @ 138 PSIA (85% HP)  
 3200 ICFM @ 120 PSIA (67% HP)  
 @ 660 HP

DISCHARGE PRESSURE, PSIA

160  
150  
140  
130  
120  
110  
100



7000 BHP 2.760 2.000

15%

Power, HP

100  
80  
60



INLET VOLUME FLOW, ICFM

2700

2800

2900

3000

3100

3200

TEST DATA CORRECTED TO SEA LEVEL CONDITIONS  
8211523B-1 TEST DATE 7/14/63 N-3570R100  
P=12.2 PSIA. R.M. 120 PSIA. R.M. 2000  
T.W. 10000 T.W. 10500  
DEPT. OF WATER & POWER  
FOR INTERMOUNTAIN POWER AGENCY

NOMINAL PERFORMANCE  
8100 ICFM @ 138 PSIA (PSI) @ 675 HP  
3200 ICFM @ 120 PSIA (PSI) @ 465 HP

Power, HP

DISCHARGE PRESSURE, PSIA

INLET VOLUME FLOW, ICFM

2500 3000 3500 4000 4500 5000 5500 6000 6500 7000 7500 8000 8500 9000 9500 10000

1000

1500

2000

3000

4000

5000

6000

7000

8000

9000

10000

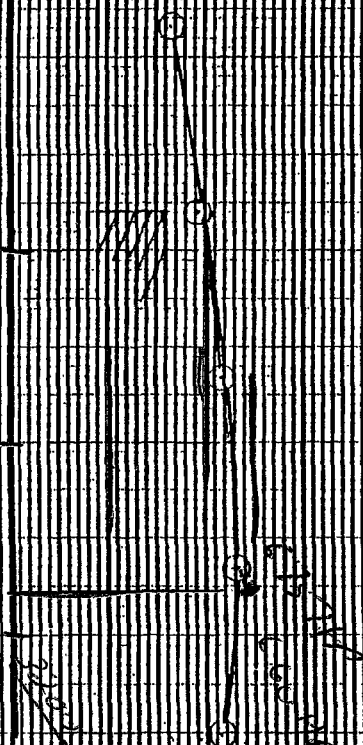
11000

12000

13000

14000

15000



INLET VOLUME FLOW = 2950 ICFM

675 HP

1000

8211523B-1 TEST DATE

IP12\_006268

TEST DATA CORRECTED TO JOB SITE CONDITIONS  
 B211523B-3 TEST DATE 7/20/83 N=3570rpm  
 $P_B = 12.2 \text{ PSIA}$   $P_{IN} = 12.0 \text{ PSIA}$   $RH = 20\%$   
 $T_{IN} = 100^\circ\text{F}$   $T_N = 105^\circ\text{F}$   
 DEPT. OF WATER POWER OF THE CITY OF L.A.  
 FOR: INTERMOUNTAIN POWER AGENCY

NOMINAL PERFORMANCE  
 3100 ICFM @ 138 PSIA (PSA)  
 @ 675 HP  
 3200 ICFM @ 120 PSIA (PSA)  
 660 HP

DISCHARGE PRESSURE, PSIA

140  
130  
120  
110  
100  
90  
80

TURN DOWNS 2850 ICFM  
 11%

Power, HP

700  
650  
600  
550  
500

2800 2900 3000 3100 3200

INLET VOLUME FLOW, ICFM

B211 523-3 TEST DATE 7/20/83

... 7th SHIP ... 7/19/83

IP12\_006269

46 1322

16 1/2" X 10 1/8" TO 15 INCH 3 X 10 INCHES  
 K&E PUMP & MOTOR CO. MADE IN U.S.A.

TEST DATA CORRECTED TO JOB SITE COND'S.  
 32115238-4 TEST DATE 7/22/83 N=3570 RPM  
 PG=12.2 PSIA P<sub>1</sub>=12.0 PSIA R.H. 20%  
 T<sub>IN</sub>=100°F T<sub>W</sub>=105°F  
 DEPT. OF WATER & POWER OF THE CITY OF L.A.  
 FOR INTERMOUNTAIN POWER AGENCY

NOMINAL PERFORMANCE  
 3100 @ 138 PSIA C  
 3200 @ 120 PSIA C

P<sub>2</sub>(2) @ 655 HP  
 P<sub>2</sub>(1) @ 660 HP

DISCHARGE PRESSURE, PSIA

160  
150  
140  
130  
120  
110

TRIMMED 2855 RPM

11%

Power, HP

700  
650  
600

2800 2900 3000 3100 3200

INLET VOLUME FLOW, ICFM

46 1322

K&E 10 X 10 TO 14 INCH 7 X 10 INCHES  
 K&E K&E & S&S CO. MADE IN U.S.A.



# Compressor-Pump & Service, Inc.

## Salt Lake City

3333 West 2400 South  
Salt Lake City, UT 84119  
Ph: (801)973-0154  
Fax: (801)973-9546

## Nevada

Elko, NV  
Ph: (775)738-2929  
Fax: (775)738-1607

## Boise

7480 Lemhi St.  
Boise, ID 83709  
Ph: (208)377-0225  
Fax: (208)377-9477

## Idaho Falls

PMB #154 / 1795 W. Broadway  
Idaho Falls, ID 83402  
Ph: (208)522-0180  
Fax: (208)522-0181

## QUOTATION

TO: Intermountain Power Service Corp.  
850 West Brush Wellman Road  
Delta, UT 84624-9546  
Attn.: Bret Kent, Engineering Manager  
Phone: (435) 864-6447 e-mail: Bret-K@ipsc.com

DATE: February 26, 2008  
QUOTE # KS-1675 Rev. D  
REF: Air Drying system replacement  
**Budget Estimate Only**  
**Includes Filter Element Costs**

| ITEM | QTY | DESCRIPTION   | PRICE EACH      | TOTAL        |
|------|-----|---|-----------------|--------------|
|      |     | <b>Air Dryers for FS Elliott PAP Centrifugal Compressor Model 310DA3, S/N B 211523-1 thru 4</b>   |                 |              |
| A    | 4   | <p>Zeks Model 2100ZBA40, 4" Flanged connection, dual tower, Blower Purge heated regenerative type air dryer. <b>Rated to deliver a -40° F pressure Dewpoint to 2242 SCFM @ 105 PSIG, with a 0% purge rate, based upon a 95 Degrees F ambient and 100% Relative Humidity entering the air dryer. Wired for 460V/3Ph/60Hz. Complete with standard equipment, including the following:</b></p> <ul style="list-style-type: none"><li>• <b>NEMA 4</b> rated panel, wired for 460V/3Ph/60Hz.</li><li>• 45 KW external heater with <b>AccuTemp solid state heater control</b>, High temperature alarm with interlock and heater fault alarm.</li><li>• <b>The blower is a 20 Hp. rated Regenerative type unit, rated for service at jobsite altitude</b></li><li>• 8 Hour Cycle, 4 hours drying, 4 hours regenerating. Please note that these times are "changed" when operating with the "Moisture Load Control" or "Power Saver" options.</li><li>• Failure to Shift alarm.</li><li>• <b>Bi-Mode operation</b>, allows the unit to operate as a conventional heatless type air dryer in the event of a heater or blower failure.</li><li>• <b>DPC programmable controller</b> with integral keypad.</li><li>• <b>DynOptic</b> indicating panel.</li><li>• <b>Compressed Air Cool Down</b>, uses 8% purge air for 1 hour</li><li>• <b>Desiccant shipped loose, for installation at the site, by others.</b></li><li>• <b>5 year warranty on flow valves and heaters</b></li></ul> <p>Approximate Weight: 7030 LBS. Each<br/>Size: 84" wide x 60" deep x 92" tall</p> | \$45,526.00 Ea. | \$182,104.00 |

IP12\_006271

| <u>Optional Equipment ADDERS:</u>        |    |   |                 |             |
|--|----|---|-----------------|-------------|
| B  | 4  | Moisture Load Control with; Digital Dewpoint display with a "High Humidity" alarm, can save up to 50% of the purge rate.  | \$2,349.00 Ea.  | \$9,396.00  |
| <u>Alternate:</u>                        |    |   |                 |             |
| C  | 4  | Power Saver, can save up to 50% of the purge rate.  | \$741.00 Ea.    | \$2,964.00  |
| D  | 4  | High Outlet Temperature Alarm   | \$504.00 Ea.    | \$2,016.00  |
| E  | 4  | Indoor Tower Insulation   | \$1,955.00 Ea.  | \$7,820.00  |
| <u>New Filters, supplied separately:</u> |    |   |                 |             |
| F  | 4  | ZEKS Model ZFF2400H, coalescing filter, .01 micron rating. Complete with auto drain valve and differential pressure indicator.  | \$3,953.00 Ea.  | \$15,812.00 |
| G  | 16 | ZEKS Part# EF600H, replacement elements for a Model ZFF2400H Coalescing Filter. <b>Note this housing requires 4 elements.</b>   | \$235.00 Ea.    | \$3,760.00  |
| H  | 4  | ZEKS Model ZFFHT2400G, High Temperature rated particulate filter, 1 micron rating with 2.0 PSID (saturated). Complete with differential pressure indicator.                                 | \$4,211.00 Ea.  | \$16,844.00 |
| I  | 16 | ZEKS Part # EFHT600G, replacement element for a Model ZFFHT2400G Coalescing Filter. <b>Note this housing requires 4 elements.</b>   | \$330.00 Ea.    | \$5,280.00  |
| <u>Filter Package, Pre-Piped:</u>        |    |   |                 |             |
| J  | 4  | "Filter Package C", with the above separate filters mounted on the air dryer with a 3 valve bypass around each of the filters. This adder would be in lieu of the filters priced separately | \$13,101.00 Ea. | \$52,404.00 |

| <b><u>Option for Mist Eliminator instead of Standard Coalescing Filter:</u></b> |   |  |                |             |
|---|---|--|----------------|-------------|
| K   | 4 | ZEKS Model 2400HDF, Mist Eliminator type coalescing filter, 0.1 micron rating and less than 1 PSID pressure Drop. Complete with differential pressure indicator.<br><b>Element life is rated at 10-15 years.</b> | \$5,981.00 Ea. | \$23,924.00 |
| L   | 4 | ZEKS Part# M2400, replacement elements for a Model 2400HDF Mist Eliminator Coalescing Filter.  | \$2,045.25 Ea. | \$8,181.00  |
| M   | * | ZEKS Model NCC1701-D, ½" NPT, <b>Electric Zero Loss Drain Valve</b> , wired for 115V/1Ph/60Hz.<br>* Quantity is subject to final design of compressed air system change.   | \$164.70 Ea.   |             |

This quotation subject to our standard conditions of sale. Quotation good for 30 days.  
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The purchase price is payable within 30 days after delivery unless specifically otherwise provided herein. The prices specified are in U.S. currency, free of all expenses to seller for collection charges. Pro rata payments shall be made for partial shipments. When in the opinion of seller the financial condition of the purchaser renders it necessary, seller may require cash payment or satisfactory security before shipment. Interest at 18% per annum or at the highest rate permitted by state law, whichever is smaller, will apply to all amounts not paid when due.

Purchaser shall pay to seller, in addition to the purchase price, in the amount of all sales, use, privilege, occupation, excise or other taxes, federal, state, local or foreign which seller may be required to pay in connection with furnishing goods or services to the purchaser.

Seller reserves the right to correct any typographical or clerical errors. All goods shall be installed at the expense of the purchaser unless specifically provided otherwise. Seller assumes no responsibility for improper operation of equipment due to faulty erection or installation by others.

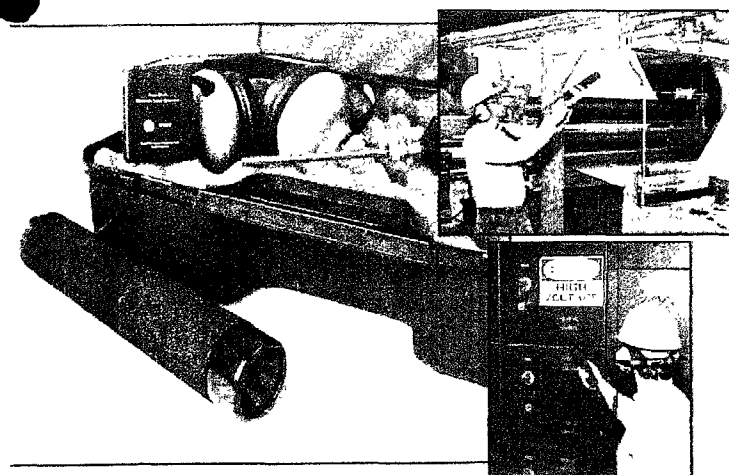
Seller shall not be liable for any loss or damage due to "force majeure," which shall be deemed to mean all causes whatsoever not reasonably within the direct control of seller, including but not limited to acts of God, war riot or insurrection, blockages, embargoes, sabotage, epidemics, fires, strikes, lockouts, or other industrial disturbances, delays of carriers, and inability to secure materials, labor or facilities.

Purchaser agrees to pay expenses of collection, including reasonable attorney fees, in the event of default.

This agreement is deemed to be entered into and is not performed at seller's place of business.



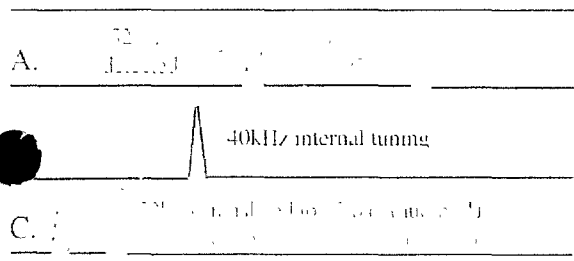
# Model VPX-WR



You might be impressed with the fact that AccuTrak® can actually hear the blink of a human eye, but what will *really* impress you is AccuTrak®'s ability to identify the hiss of a compressed air leak, even in the noisiest facility.

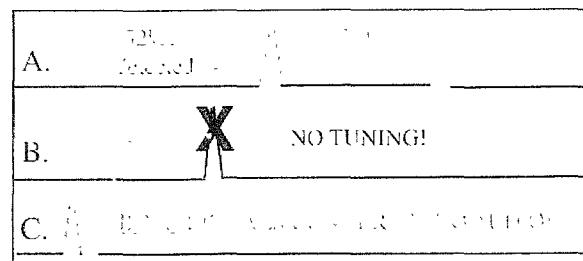
AccuTrak® **Model VPX-WR** represents *THE* state-of-the-art technology for ultrasonic air leak detection! This *ultra-durable and waterproof* instrument incorporates our *patented and trademarked* circuitry we call "DND" (Dynamic Noise Discrimination). Older detectors using only the heterodyne technique, simply cannot offer the same leak detection abilities, especially in loud plant environments.

Fixed-tuning type detectors compare the incoming signal (A) with an internally generated frequency (B), in most cases around 40 kHz.



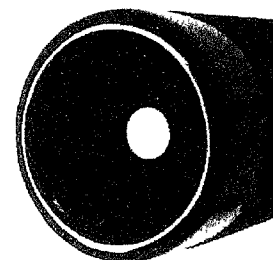
The difference between these two signals is what gets heard in the headset (C). One problem (as seen here) is that if the *incoming* sound is too far from the *internally generated* frequency the difference is too large (high in frequency) to ever be heard by the user. "Fixed-tuning" systems do not widen the frequency detection range of the instrument, only lock in on one internally generated one.

AccuTrak® constantly "scans" the complete range of detection from 20 to 100kHz and automatically tracks both frequency and intensity of

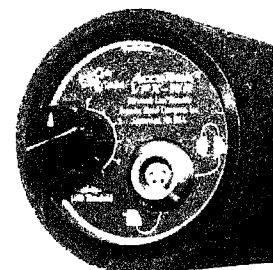


incoming signals (A). When a sound is detected which is above the previous intensity baseline, it uses a sophisticated mapping process rather than simple subtraction to translate the sound to the audible range (C). The result is a leak detector able to find leaks faster than any other; *no tuning!* You can see from the example above that DND captures both incoming signals while standard heterodyning could hear only one.

## Waterproof Sensor!



## Single Control!



## Find leaks fast! Find leaks fast!

Leaks come in all sizes, shapes, frequencies and intensities. You don't care what frequency a leak sound is, you just want to know where the leak is! Since the intensity of the leak sound is not directly proportional to the size of the leak, you can't use a meter to quantify CFM or volume of air lost, but you CAN find it! - THAT is what the VPX-WR does and does well... Finds leaks fast!

## Find leaks fast! Find leaks fast! Find leaks fast!

AccuTrak® VPX-WR has the only sensor in the world that combines high sensitivity and hermeticity. Its resistance to water, steam, oils, fumes, dust, etc. allows the instrument to be used in hostile environments where other detectors fail. Totally sealed controls, indicators, and plugs make the unit 100% water tight. The internal circuitry is housed in hard anodized aluminum for durability and longevity. The aluminum housing is surrounded with a padded grip for comfort and easy handling. It is the most durable instrument of its kind.

|                                      |   |   |
|--------------------------------------|---|---|
|                                      |   |   |
| <b>Physical</b>                      |   |   |
| Size                                 | Cylindrical, 1.75' O.D.   | <b>AccuTrak CAPK</b><br>(Compressed Air Package Kit)<br>Includes:<br>VPX-WR Ultrasonic Detector<br>Sealed Metal Body w/<br>Padded Grip<br>Deluxe Stereo Head Set<br>Touchprobe<br>Wave Guide<br>Horn Restrictor<br>Carrying Case<br>Battery Charger - AC Plug<br>Instructional CD   |
| Weight                               | 14 oz.  |   |
| Construction                         | Anodized Aluminum   |   |
| Sensor                               | Piezoelectric, totally waterproof   |   |
| Displays                             | N/A   |   |
| Headset                              | Deluxe military style, for use with hardhat   |   |
| Carrying Case                        | Injection molded with foam inserts  |   |
| Standard Accessories                 | Standard contact probe, airborne waveguide, battery charger, optional output cable. |   |
| <b>Performance</b>                   |   |   |
| Sensitivity                          | -75dB/v/μbar  | <b>Distributor Net Price</b>  |
| Frequency Range<br>Contact -         | 20kHz to 100kHz   |   |
| Airborne -                           | 20kHz to 100kHz   |   |
| Frequency Selection                  | Automatic "DND"   |   |
| Frequency Conversion                 | "DND" and Heterodyne mix  | <b>VanHala Industrial Inc.</b><br><b>14812 Detroit Ave.</b><br><b>Cleveland OH 44107</b><br><b>216/521-6283</b><br>evanhala@yahoo.com   |
| Adjustable Parameters                | Sensitivity / Threshold   |   |
| <b>Power</b>                         |   |   |
| Battery Type                         | Re-Chargable NiMH   |   |
| Battery Life (avg. charge)           | 8 hrs. continuous   |   |
| Avg. Charge Time                     | 16 hrs.   | <div><div><div>Warranty</div><div>AccuTrak<sup>®</sup> is warranted for one year to be free of manufacturing defects, with the first 30 days unconditional! If you don't think AccuTrak<sup>®</sup> is the most accurate, and effective ultrasonic instrument available, simply return it (in good condition of course) for a full credit or refund!</div></div><div>AccuTrak<sup>®</sup> detectors are covered by numerous patents, and patents pending.</div></div> |
| Low Batt. Indicator                  | Flashing LED  |   |
| <b>Sound Generator<br/>Optional)</b> |   |   |
| Dimensions                           | 4.16" x 2.4" x .866"  |   |
| Construction                         | Durable ABS   |   |
| Indicators                           | Red LED   |   |
| Frequency Control                    | Precision Crystal Oscillator  |   |
| Output Frequency                     | 40 kHz (+/- 2.5 Hz)   |   |
| Output Intensity                     | 115 dB at 30cm. (nominal)   |   |
| Dual Mode Output                     | Continuous or Burst tone  |   |
| Power                                | 9 volt cell   |   |
| Battery Life (approx.)               | 70 hrs. Continuous, 90 hrs. Burst   |   |

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Compressors

# Control the Blow

Use intake  
throttling to  
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compressors



*By Hank van Ormer and Scott van Ormer*

**T**he concept of inlet throttling is subject to differing opinions and there is limited public test data available to choose between them. A practical look at inlet butterfly valves and inlet guide vanes for capacity control on constant-speed centrifugal compressors starts with the terminology.

## Compressor load

The loads on any air compressor are system frictional resistance, piping backpressure and the head the load imposes. Centrifugal compressors operate at a discharge pressure the system imposes on them. System pressure varies with the air used and the system's pressure drop. Resistance loads in most compressed air systems consist of multiple demands with some additional frictional pressure loss. Basic regulation normally uses

constant speed and regulates to discharge pressure. A flow reduction produces a corresponding power consumption reduction. A variable-speed control on a centrifugal compressor generally isn't effective with this type load because pressure varies as the square of the speed.

## Power

An electric bill is based on kilowatts and time. At full load, the capacity or unloading control has no effect on that bill. At part load, the capacity control translates reduced air demand into reduced energy input. Proper selection, application and maintenance of these controls help minimize energy costs.

Power is proportional to the head times the mass flow divided by the stage efficiency, plus mechanical

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## Compressors

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cent. Most engineers cite an average overall enhancement under real world operating conditions of 2 percent to 3 percent with a maximum of 4 percent.

To determine the actual advantage of inlet guide vanes over inlet butterfly valves in recovered electrical (or other) energy cost, determine the duration of turndown. Measure the actual kW across the turndown range. Deduct the appropriate percent from this total kW. Then multiply by the number of operating hours and the power rate to determine the projected energy cost. When evaluating a new machine, go through the same exercise using the machine-specific performance curve.

### **Inlet guide vanes and idle power draw**

The idle power draw is a function of the mass flow of the air and the discharge pressure at each stage. This appears to be controlled by the manufacturer's desire to maintain a minimum pressure.

There is no question that some inlet guide vanes don't seal as completely as a high-quality, full-seat butterfly valve. But the technology exists for guide vanes to seal more than enough for any required minimum flow. The inlet guide vane's turndown operational flow is more stable than the inlet butterfly valve's.

### **Valve vs. vane recap**

Other than not being able to reach total closure, there's no reason that inlet guide vanes can't be manufactured to hold the same idle power draw as the inlet butterfly valve on

**Selecting an inlet guide vane control to take advantage of its over-throttle or counter-rotation operating characteristics can provide as much as 20 percent more flow if the motor horsepower is available.**

most machines. Designers seem to agree on an average idle draw of 20 percent with inlet butterfly valves and perhaps as high as 30 percent with inlet guide vanes.

Determining the initial cost and energy savings from inlet guide vanes is relatively easy. But if you use them, there are some important points to remember:

- Selecting an inlet guide vane control to take advantage of its over-throttle or counter-rotation operating characteristics can provide as much as 20 percent more flow if the motor horsepower is available. You'll give up a little efficiency at full load, but can throttle back into a high-efficiency range.
- Inlet guide vanes are excellent when system resistance is primarily frictional. Vanes tend to follow the system resistance load better than inlet butterfly valves and are better at avoiding blow off.
- Inlet guide vanes offer a stable turndown, which often allows better use of full turndown when compared to the higher turbulent flow from an inlet butterfly valve.
- Building in a high rise-to-surge

will force the unit to operate down the curve in a less efficient region. Good throttle range and high efficiency at design are difficult to achieve simultaneously. Inlet guide vanes allow wider throttle range at higher efficiencies than inlet butterfly valves.

- With the increased turndown efficiency, there is more air flow at the same brake horsepower, and during colder weather significantly more air is available (as much as 20 percent) if the motor horsepower is available.
- Under certain conditions this can be important, particularly if the extra air flow allows shutting down a unit that would otherwise be operating, particularly at part load.
- Inlet guide vanes vary in shape, material and drive mechanism. One type of control may be more precise than another or operate better than others under certain conditions. ®

*Hank van Ormer and Scott van Ormer own and operate AirPower USA, Inc. They can be reached at [HankvanOrmer@aol.com](mailto:HankvanOrmer@aol.com) or 740-862-4112.*

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## Compressors

shift from being parallel to the air stream to fully perpendicular. The effect is to reduce the work required to produce the same air discharge condition. The net result is reduced input power and improved specific power at low flow. Inlet guide vanes also can increase the flow when oriented in the over-throttle position (flow against rotation). There appears to be no fixed amount of flow gain, but it is estimated to be as high as 20 percent. This increased flow requires commensurate additional horsepower.

### Inlet guide vane performance

Inlet guide vanes reduce the power required to produce a lower-than-design flow at the same pressure more than an inlet butterfly valve can reduce it. Simply put, inlet guide vanes are more efficient at turndown control than butterfly valves. Basic guide vane facts include:

- The efficiency improvement from having inlet guide vanes only on the first stage decreases as the number of compression stages increase.
- The better the inlet guide vanes are adjusted, the greater their impact on performance.
- Other than over-throttle flow increase, inlet guide vanes offer no benefit at full load.
- Inlet guide vanes should be mounted no more than one pipe diameter away from the inlet, but not so close that the vane swirl sets up harmonics with the impeller's vane pass frequency.

- Although inlet guide vanes don't increase turndown, they do enhance efficiency during turndown.

### Inlet guide vane benefits

Figure 2 shows typical performance curves for inlet guide vanes and an inlet butterfly valve. Note the 4.2 percent to 8 percent difference in specific power. The greatest improvement is at the fully closed position. Many factors affect this fully unloaded value—not the least being the minimum mass flow. The actual power at idle with inlet guide vanes will probably be no lower than 20 percent at full load and can rise as high as 30 percent.

Figure 3 reflects performance at various ambient conditions. The design flow and discharge is always stated at the worst case conditions—high-temperature air and warm cooling water. For example, extreme summer conditions (air at 95° F and 60 percent RH, water at 85° F) maximizes the work required to compress air. The point is that the compressor runs at these conditions only for a limited time. An accurate benefit evaluation should include performance in colder weather and on average days.

These curves show inlet guide vanes perform from 3 percent to 6 percent better in hot design conditions. At lower temperatures, better performance also can be expected. The basic performance at colder conditions (air at 30° F, water at 70° F) now goes from 5.2 percent to 8 percent. On an average day with air at 80° F and water at 80° F, the gain is from 4.2 percent to 8 per-

uses about 20 percent power at idle.

- Well applied, high-quality inlet butterfly valves have a much better ability to stop the air flow than most inlet guide vanes. The inlet butterfly valve is predictable, and its performance is predictable and repeatable.
- Some compressors may require more mass flow to control or eliminate thrust loads that could damage critical components. These units have a higher power draw at idle.

#### **Inlet butterfly valve recap**

The inlet butterfly valve reduces the centrifugal compressor's power requirement in relation to reduced airflow during part-load conditions while maintaining the design discharge pressure. Because the pressure always decreases across the butterfly valve, it can do nothing to improve compression efficiency or specific power.

Many variables, including the machine and its location, determine idle power. However, good judgment indicates that if you are to run at full idle for long stretches, check your kW. If this unit has an inlet butterfly valve and is drawing more than 20 percent of full power, find out why. Review the factors that affect power draw at idle. Consider an electronic control system to monitor and control the compressor.

#### **Inlet guide vanes**

Inlet guide vanes are usually mounted on the compressor's first stage inlet, but are often installed on each stage in larger

process units. Like butterfly valves, inlet guide vanes vary the volumetric flow at constant discharge.

shift from being parallel to the air stream to fully perpendicular. The effect is to reduce the work required to produce the same air discharge condition. The net result is reduced input power and improved specific power at low flow. Inlet guide vanes also can increase the flow when oriented in the over-throttle position (flow against rotation). There appears to be no fixed amount of flow gain, but it is estimated to be as high as 20 percent. This increased flow requires commensurate additional horsepower.

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Inlet guide vanes produce a swirl in the airflow, usually in the direction of impeller rotation. When throttling flow, the vanes

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#### **Inlet guide vane benefits**

Figure 2 shows typical performance curves for inlet guide vanes and an inlet butterfly valve. Note the 4.2 percent to 8 percent difference in specific power. The greatest improvement is at the fully closed position. Many factors affect this fully unloaded value—not the least being the minimum mass flow. The actual power at idle with inlet guide vanes will probably be no lower than 20 percent at full load and can rise as high as 30 percent.

Figure 3 reflects performance at various ambient conditions. The design flow and discharge is always stated at the worst case conditions—high-temperature air and warm cooling water. For example, extreme summer conditions (air at 95° F and 60 percent RH, water at 85° F) maximizes the work required to compress air. The point is that the compressor runs at these conditions only for a limited time. An accurate benefit evaluation should include performance in colder weather and on average days.

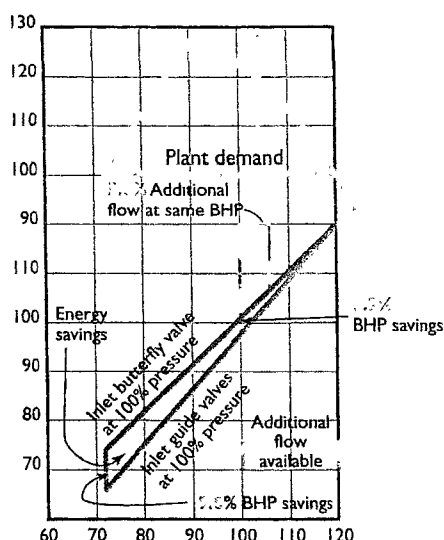
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## Compressors

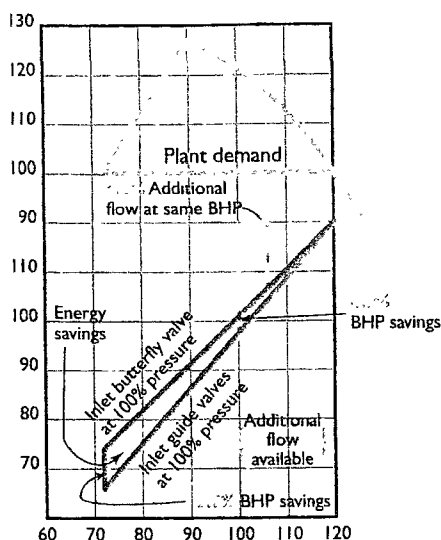
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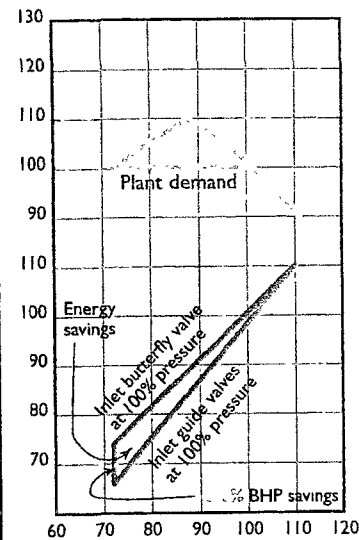
### Inlet conditions

Barometric pressure 14.1 psia  
 Inlet pressure 14.1 psia  
 Inlet temperature 30°F  
 Relative humidity 60%  
 Chilled water temperature 70°F  
 Gas Air



### Inlet conditions

Barometric pressure 14.4 psia  
 Inlet pressure 14.1 psia  
 Inlet temperature 50°F  
 Relative humidity 80%  
 Chilled water temperature 80°F  
 Gas Air



### Inlet conditions

Barometric pressure 14.4 psia  
 Inlet pressure 14.1 psia  
 Inlet temperature 95°F  
 Relative humidity 60%  
 Chilled water temperature 85°F  
 Gas Air

Figure 3. Energy savings of inlet guide vanes over butterfly valves at partial loads are greater at lower inlet ambient and chilled water temperatures. (Joy Mfg. Co.)

At a predetermined discharge pressure short of the surge point, the compressor unloads as the inlet valve closes and the bypass or unloading valve opens.

On most equipment, the inlet throttling valve can be controlled to a very narrow band to achieve a constant pressure.

So, what's the idle power input with a properly applied throttle valve? The most common answer is a high of 30 percent with an average of 20 percent, although sometimes it can be as low as 15 percent.

Before you ask if that's as good as it gets, consider the following:

- The mechanical fixed power

draw is not an industry average. It's specific to a frame size.

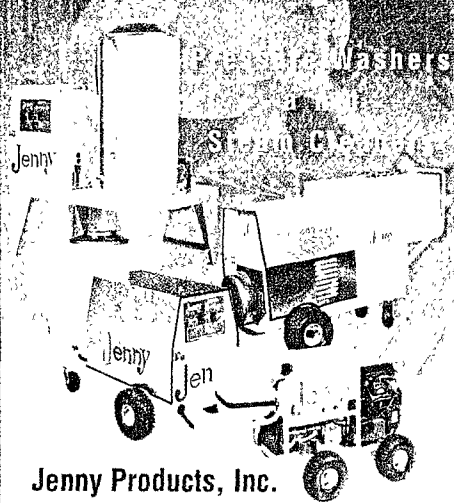
- The mass flow of air at closure—and thus the power—varies as the inlet conditions change. Without adjustment, it'll be higher in colder weather and lower when a storm approaches.
- Backpressure on the unloader or blowoff valve is significant.
- A maladjusted control linkage that leaves the valve more open than necessary passes more mass flow and consumes additional power.
- Most compressors require a minimum flow and pressure at idle for cooling and to avoid pulling a vacuum on the first stage,

which could pull oil from the drive train into the compressor. Most designs avoid this by blowing high-pressure air towards the gear case along the pinion shaft and through the seal.

- At least one manufacturer manifolds the discharge of all stages at idle. Even if the first stage pulls a vacuum, the positive pressure in the following stages buffer the vacuum to preclude oil migration. This allows the first stage to start at a lower pressure and, therefore, requires a potentially lower base power draw at full idle. However, with a good inlet butterfly drive, it still

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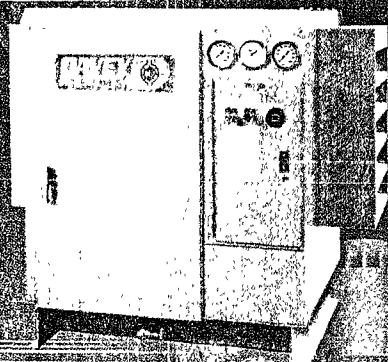


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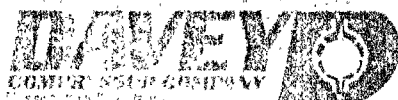
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## Compressors

### Valve vs. vane

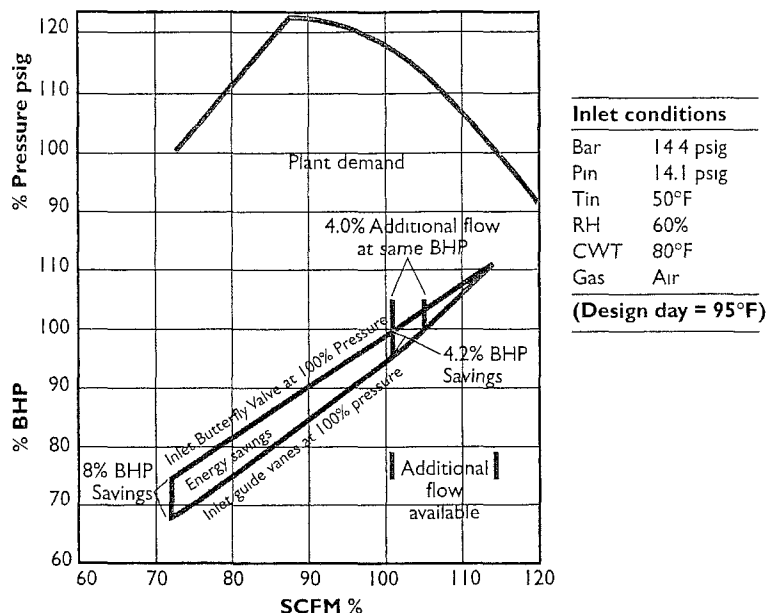


Figure 2. Inlet guide vanes are more efficient at partial loads. (Cooper Turbo, Inc.)

Other inlet butterfly valves use a full seating design with a machined opening for bypass air.

Regardless of terminology, the power the compressor requires at idle is a function of the mass flow and the discharge pressure at each stage. The mass flow thus is a function of the pressure at the entrance to the first stage.

Most designers believe a partly open butterfly valve that's controlled by stop adjustment is not efficient. They argue the margin for error is too great, and the turbulent flow around the valve may cause unpredictable results. There's no doubt that restricted flow won't fill the impeller as effectively as would be the case in the wide-open position; thus, the specific power decreases.

Regardless of type, the inlet butterfly valve is applied in several ways:

- Constant pressure or baseload: This design matches compressor output to the demand. As the pressure rises, the bypass valve opens to vent excess air to atmosphere. There is no reduction in power draw. This form of base load control is considered obsolete for variable loads.
- Inlet throttle or modulation control: The inlet butterfly forces the compressor to operate on its characteristic curve when the demand is less than rated capacity. As the inlet valve closes and the rising pressure approaches surge point, the bypass valve matches demand by opening slowly to bleed off excess air. Once the bypass valve is fully open and/or the inlet valve is at full turndown adjustment, there's no further reduction in power.
- Full unload (auto dual) control:



## Compressors

**AIR POWER, USA • P.O. BOX 292**  
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losses realized in the compression cycle. Most centrifugal compressor manufacturers use the following definitions.

**Brake horsepower** is the power input at the compressor shaft needed to compress the air. At least one manufacturer excludes mechanical losses.

**Shaft horsepower** refers to the power input at the compressor shaft to compress the air and includes mechanical losses, which are machine-specific.

**Input power** is the shaft horsepower plus mechanical and electrical losses in the drive system. This is the power that determines the electric bill.

The power required for a centrifugal compressor is a function of the mass flow rate of air and the discharge pressure. This implies that:

- Higher inlet pressure means more mass flow and more power required.
- Colder inlet air means more mass flow and more power required.
- Lower temperature cooling water means more mass flow and more power required.

### Surge

Each centrifugal compressor has a specific discharge pressure at which the air becomes turbulent at the impeller tips, causing the phenomenon of surge. This increased turndown is referred to as rise-to-surge. Other conditions affect the action of a constant-speed centrifugal capacity control. For example:

- As the discharge pressure rises, the rated inlet cfm falls.
- As the pressure rises, the surge

point shifts upward to a higher percentage of flow.

Figure 1 illustrates that for a particular compressor operating at its design pressure, the surge point is about 65 percent of flow. When the unit is pushed to 118 percent of design pressure, the surge point rises to 75 percent of rated inlet cfm, which makes control less stable.

### Inlet butterfly valve

From the late 1970s to the early 1980s, the inlet butterfly valve was the control of choice for industrial multi-stage centrifugal compressors. Mounted on or near the first stage inlet, the valve closes in reaction to a rise in system pressure. The falling pressure on its downstream side is the inlet pressure at the impeller and diffuser. As the pressure drop across the valve increases, the density of the entering air decreases. This results in a lower mass flow in relation to inlet ambient cfm. Power draw falls, but

not proportionally to the decrease in mass flow. This implies the specific power (scfm per input kW) falls. Additionally, as the butterfly valve reaches the end of its closure, it produces turbulence, which further reduces the effective flow into the impeller.

At full idle, the inlet butterfly valve closes, and the inlet bypass valve or unloader valve opens. Theoretically, the compressor is now moving just enough air for cooling, avoiding vacuum and minimizing the power draw.

Ability to control this flow precisely under varying inlet conditions is a function of the specific equipment or valves used, as well as how they are adjusted and maintained. Some inlet butterfly valves are non-seating with actuator stops at full open and full close. But the term "full close" is misleading. Air must bleed around valve to flow through the unit to eliminate first-stage vacuum while minimizing mass flow and discharge pressure.

### Surge happens

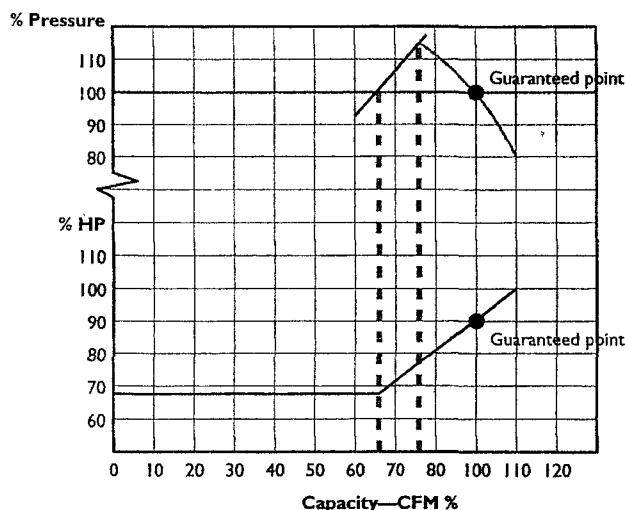


Figure 1. At about 118% of design pressure the surge point rises to 75% of flow, making control less stable. (Joy Mfg. Co.)

# The secret is in the pipe

By Hank van Ormer, Don van Ormer and Scott van Ormer

THERE'S NO SUCH THING AS TOO  
LARGE A COMPRESSED AIR LINE

A common error we see in compressed air systems, in addition to poor piping practice, is line sizes too small for the desired air flow. This isn't limited to the interconnecting piping from compressor discharge to dryer to header. It also applies to the distribution lines conveying air to production areas and within the equipment found there. Undersized piping restricts the flow and reduces the discharge pressure, thereby robbing the user of expensive compressed air power. Small piping exacerbates poor piping practices by increasing velocity- and turbulence-induced backpressure. (See "There's a Gremlin in your air system — Its name is turbulence," *Plant Services*, July 2002, p. 37).

Pipe size and layout design are the most important variables in moving air from the compressor to the point of use. Poor systems not only consume significant energy dollars, but also degrade productivity and quality. How does one properly size compressed air piping for the job at hand? You could ask the

pipefitter, but the answer probably will be, "What we always do," and often that's way off base.

Another approach is matching the discharge connection of the upstream piece of equipment (filter, dryer, regulator or compressor). Well, a 150-hp, two-stage, reciprocating, double-acting, water-cooled compressor delivers about 750 cfm at 100 psig through a 6-in. port. But most 150-hp rotary-screw compressors, on the other hand, deliver the same volume and pressure through a 2-in. or 3-in. connection. So, which one is right? It's obvious which is cheaper, but port size isn't a good guide to pipe size.

## Charts and graphs

Many people use charts that show the so-called standard pressure drop as a function of pipe size and fittings, which sizes the line for the what is referred to as an acceptable pressure drop. This practice, too, can be misleading because the charts can't

accommodate velocity- and flow-induced turbulence. Think about it. According to the charts, a short run of small-bore pipe exhibits a low total frictional pressure drop, but the high velocity causes an extremely large, turbulence-driven pressure drop. Then there's the question of the meaning of acceptable pressure drop. The answer to this question often isn't supported by data, such as the plant's electric power cost to produce an additional psig.

We've audited many plants during the past 20 years and found the unit cost of air for positive-displacement compressors runs from several hundred dollars per psig per year to several thousand dollars per psig per year. At current energy costs, you don't want the pipe to be a source of pressure drop.

### Shooting blind

Not knowing the energy cost of lost pressure as a function of line size can lead to a blind decision. Unfortunately, this is what we find in most of the air piping systems installed during the past 30 years. Older systems that were designed with care are often right on the mark, except if they've been modified after the original installation.

Some might call pipe sizing a lost art, but we see the issue as a lack of attention to detail, basic piping principles and guidelines. Read on to learn how to size air piping using velocity, which, when combined with appropriate piping practice, ensures an efficient compressed-air distribution system. As compressed-air system consultants and troubleshooters, we use these guidelines to design new piping systems and to analyze existing system performance and opportunities for improvement.

### Interconnects and headers

The interconnecting piping is a critical element that must deliver air to the distribution headers with little pressure loss, if any. This isn't only an energy question. It also ensures the capacity controls will have sufficient effective storage to allow them to react to real demand and translate less air usage to a comparable reduction in input electrical energy.

often isn't supported by data, such as the plant's electric power cost to produce an additional psig.

We've audited many plants during the past 20 years and found the unit cost of air for positive-displacement compressors runs from several hundred dollars per psig per year to several thousand dollars per psig per year. At current ener-

The main distribution headers not only move air throughout the plant, they also supply the appropriate local storage that ensures the process feeds have adequate entry pressure and flow. The main header system represents storage that supports the operating pressure band for capacity control. You want the pressure drop between compressor discharge and point of use to be significantly less than the normal operating control band (10 psig maximum).

### The targets

The objective in sizing interconnecting piping is to transport the maximum expected volumetric flow from the compressor discharge through the dryers, filters and receivers to the main distribution header with minimum pressure drop. Contemporary designs that consider the true cost of compressed air target a total pressure drop of less than 3 psi.

Beyond this point, the objective for the main header is to transport the maximum anticipated flow to the production area and provide an acceptable supply volume for drops or feeder lines. Again, modern designs consider an acceptable header pressure drop to be 0 psi.

Finally, for the drops or feeder lines, the objective is to deliver the maximum anticipated flow to the work station or process with minimum or no pressure loss. Again, the line size should be sized for near-zero loss. Of course, the controls, regulators, actuators and air motors at the work station or process have requirements for minimum inlet pressure to be able to perform their functions. In many plants, the size of the line feeding a work station often is selected by people who don't know the flow demand and aren't aware of how to size piping.

In our opinion, new air-system piping should be sized according to these guidelines. For a system that doesn't meet the criteria, the cost of modification must be weighed against the energy savings and any improvements in productivity and quality.

Obviously, the lower the pressure drop in transporting air, the lower the system's energy input. Lower header pressure also reduces unregulated air flow (including leaks) by about 1% per psi of pressure reduction.

### Eliminate the drop

Most charts show frictional pressure drop for a given flow at constant pressure. Wall friction causes most of this loss, which is usually denominated as pressure drop per 100 ft. of pipe. Similar charts express the estimated pressure loss for fittings in terms of additional length of pipe. When added to the length of straight pipe, the sum is called total equivalent length. These charts reflect the basic calculations for pressure loss, which include:

- Air density at a given pressure and temperature.
- Flow rate.
- Velocity at pipeline conditions.
- The Reynolds number.
- Other factors, including a friction factor based on the size and type of pipe.

The calculations and chart data help to identify only the probable minimum pressure drop. Internal roughness and scaling dramatically affect the pipe's resistance to flow (friction loss). Resistance increases with time as the inner wall rusts, scales and collects more dirt. This is particularly true of black iron pipe.

Table 1. Compression ratios at gauge pressures

| psig | Compression ratio |
|------|-------------------|
| 60   | 5.05              |
| 70   | 5.76              |
| 80   | 6.44              |
| 90   | 7.12              |
| 100  | 7.8               |
| 110  | 8.48              |
| 120  | 9.16              |
| 130  | 9.84              |
| 140  | 10.52             |
| 150  | 11.20             |
| 200  | 14.5              |

ume, intermittent demand produces dramatic pressure drop during peak periods. Ignoring this fact affects every process connected to the header. For more detail, see "The compressed air receiver: The endless question," *Plant Services*, May 1997, p. 49, and Appendix 1, Tables and Outline from "DOE/CAC Air Master Training Manual." For a given size pipe:

- At constant pressure, the greater the flow, the greater the loss per foot of pipe.
- At constant flow rate, the lower the inlet pressure, the greater the loss per foot of pipe.
- At any condition, smooth-bore pipe (copper, stainless steel) exhibits lower friction losses.

### Air velocity

The most overlooked idea in piping layout and design is air velocity. Excessive velocity can be a root cause of backpressure, erratic control signals, turbulence and turbulence-driven pressure drop.

The British Compressed Air Society suggests that a velocity of 20 fps or less prevents carrying moisture and debris past drain legs and into controls. A velocity greater than 30 fps is sufficient to transport any water and debris in the air stream. Thus, the recommended design pipeline velocity for interconnecting piping and main headers is 20 fps or less, and never to exceed 30 fps. Field testing reveals that, under these conditions, air stream turbulence is generally negligible. Line drops, feed lines or branch lines less than 50 ft. long work well at a velocity of 30 fps, but velocity here should not exceed 50 fps.

### Crunching numbers

First, look at the velocity at maximum anticipated flow conditions using the following equation:

$$V = 3.056 * Q/D^2 \quad (\text{Eqn 1})$$

Where V = air velocity (ft./sec.)

Q = volumetric flow rate (cfm)

D = conduit inside diameter (inches)

Although this method of determining the minimum pipe

size on the basis of air velocity is easy, you also must consider that the compressed air volume is expressed in cubic feet per minute of free air, which is the air volume at ambient atmospheric conditions, not the compressed volume.

To adjust the inlet air volumetric flow rate to actual pipeline conditions, you'll need to divide the volume of free air by the compression ratio using the following equation:

$$CR = (P + P_a)/P_a \quad (\text{Eqn 2})$$

Where P = line pressure (psig)


P<sub>a</sub> = average atmospheric pressure at your elevation (psi)

Table 1 shows the compression ratio as a function of gauge pressure for a location at sea level, where the atmospheric pressure is 14.7 psi. At higher elevations, the average atmospheric pressure drops and the compression ratio rises. For example, Flagstaff, Ariz., at a 7,000-ft. elevation, has an average atmospheric pressure of about 11 psi. At 100 psig, the compression ratio is equal to 10 (i.e. 111/11).

To determine the pipeline velocity at conditions, merely divide the velocity given in Equation 1 by the compression ratio given in Equation 2. After selecting the minimum pipe size on the basis of velocity, check any long runs for excessive pressure drop using an appropriate drop chart. For example, a velocity of 25 fps in black iron pipe represents about 0.25 psi loss per 100 ft. of run. Although this is a little above the recommended minimum of 20 fps and, depending on the layout, would probably be acceptable from a turbulence standpoint, a high total frictional loss might dictate using a larger pipe.

This might seem to be somewhat complicated at first, but it's the most accurate way to avoid problems in sizing compressed air piping. Once you get the hang of it, it's easy to use.

**The calculations and chart data help to identify only the probable pressure drop.**

After carefully selecting a conduit size that eliminates unnecessary loss, be sure to pay the same attention to downstream items such as quick disconnects, regulators, filters, controls, fittings, number of drops from a given header and number of connections per header, so as not to offset the gains made with the pipe. Good piping performance is not an accident — it takes planning. 

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Figures: AirPower USA



# Hosing down your losses

How to specify the right hose

**A**lmost every compressed air system uses flex hose to make the final connection to production machinery. Proper selection and application of this air hose and the quick disconnects is critical to achieving optimum performance.

Compressed air system audits often uncover significant opportunity for savings at such locations. Typi-

## Never select air hose unless you know the air flow and hose length the tool requires.

cally, total system pressure is unnecessarily high to offset pressure drops in small-diameter hose and incorrect quick disconnects.

The most important sizing data for any process is the air flow and minimum pressure required at the tool entry. If you don't know these data, it's easy for system analysts to measure them on-site. In areas where the pressure or flow are critical to productivity or quality, economical mass flowmeters and pressure gauges can be rigged for continuous machine monitoring.

### Working by hand

Air-driven tools can illustrate the effect of hose and connector selection on productivity and quality. Most air tools are designed for a hose feed pressure of 90 psig. The tool designer really sizes for full flow at about 80 psig for optimum performance. Depending on the tool, pressure significantly higher than 90 psig may not increase performance, but lower pressure certainly will reduce it. In many cases, out-of-range air pressure can damage tools and reduce the time between rebuilds.

Standard impact tools, screwdrivers, grinders, chippers and banders prefer a constant 80 psig to 90 psig inlet pressure. The phrase "at rest pressure" has no meaning. Table 1, abstracted from selected air tool technical data sheets, clearly shows the general magnitude of performance loss at low pressure. At 70 psig, most tools will still operate, but below rating. At 60 psig, performance is seriously degraded and probably will be unacceptable. Operating below 60 psig isn't really a viable option. However, unless specifically stated, no tool is designed for inlet pressure greater than 100 psig. Table 2 shows the approximate performance losses at various inlet pressures in 1-hp to 3-hp vane motor grinders and sanders. The power drops

**Table 1. Performance data for air-operated tools****Typical vane air motor performance at various inlet pressures (actual results will vary by manufacturer and model)**

| Inlet air pressure (psig) |                  | ½ hp   | ¾ hp  | 1 hp  | 1½ hp | 2 hp  | 3 hp  |
|---------------------------|------------------|--------|-------|-------|-------|-------|-------|
| 60                        | rpm at max load  | 8,500  | 5,809 | 3,810 | 5,550 | 3,730 | 3,900 |
|                           | Max hp           | 0.35   | 0.47  | 0.765 | 0.927 | 1.74  | 2.32  |
|                           | scfm at max hp   | 20     | 20.1  | 27.5  | 30    | 51    | 67    |
|                           | Max torque ft-lb | 0.36   | 0.88  | 1.67  | 1.67  | 3.7   | 5.0   |
| 70                        | rpm at max load  | 9,000  | 6,184 | 4,060 | 5,900 | 3,975 | 4,160 |
|                           | Max hp           | 0.41   | 0.58  | 0.95  | 1.15  | 2.16  | 2.88  |
|                           | scfm at max hp   | 21     | 53    | 32    | 35    | 60    | 78    |
|                           | Max torque ft-lb | 0.42   | 1.0   | 1.95  | 1.95  | 4.3   | 5.8   |
| 80                        | rpm at max load  | 9,500  | 6,429 | 4,250 | 6,190 | 4,160 | 4,350 |
|                           | Max hp           | 0.5    | 0.69  | 1.13  | 1.38  | 2.58  | 3.44  |
|                           | scfm at max hp   | 22     | 27    | 36    | 40    | 68    | 89    |
|                           | Max torque ft-lb | 0.5    | 1.2   | 2.2   | 2.2   | 4.9   | 6.7   |
| 90                        | rpm at max load  | 10,000 | 6,700 | 4,400 | 6,400 | 4,300 | 4,500 |
|                           | Max hp           | 0.6    | 0.8   | 1.4   | 1.5   | 3.0   | 4.0   |
|                           | scfm at max hp   | 24     | 30    | 39    | 42    | 76    | 100   |
|                           | Max torque ft-lb | 0.55   | 1.3   | 2.5   | 2.5   | 5.5   | 7.5   |
| 100                       | rpm at max load  | 10,500 | 6,888 | 4,520 | 6,580 | 4,415 | 4,630 |
|                           | Max hp           | 0.6    | 0.9   | 1.5   | 1.8   | 3.4   | 4.6   |
|                           | scfm at max hp   | 26     | 33    | 45    | 50    | 85    | 111   |
|                           | Max torque ft-lb | 0.6    | 1.4   | 2.8   | 2.8   | 6.1   | 8.3   |

may preclude effective job performance. Along with the loss in power, which is most important, there's also a loss in speed. Both factors affect productivity.

#### Beware of ¾-in. hose

Never select air hose unless you know the air flow and hose length the tool requires. The most common hose sizes for plant use range from ¾ in. to 1 in. and handle 300 psig. Hose choice is often left to the operator, who usually wants ¾-in. hose, regardless of application, because:

- ¾-in. hose appears to be the lightest and easiest to handle.
- A 50-ft. length of ¾-in. or 1-in. hose weighs about 13 lbs., depending on grade but a 50-ft. length of ¾-in. hose weighs 22 lbs.
- The operator might not be trained regarding the hose size required to run the tool.

A ¾-in. hose isn't a viable supply hose for industrial tools. The smallest size you should use is 1 in. Table 3

refers to premium black industrial air hose. The data leads us to specific conclusions:

- ½-in. hose in 50 ft. lengths is suitable only for 1 hp or smaller tools (approximately 30 cfm/hp).
- ¾-in. hose is acceptable for 2 hp to 3 hp (60 scfm to 90 scfm), depending on the length of run.
- For runs greater than 50 ft., use larger hose or pipe, supported on the walls or ground as required, to eliminate pressure drop.
- For more comfort and easier operation, adding an 8-ft. to 10-ft. whip hose to the larger ¾-in. or 1-in. main line will have minimal effect on performance, but still gives the operator the feel of a lighter hose.
- Don't use any more hose than necessary. Coiling the extra just adds pressure drop. Cut the hose to the proper length and install fittings.

Don't forget about OSHA safety requirements. Going from a ¾-in. to 1-in. hose still allows personnel to handle

smaller hose without the mandatory automatic air shutoff valve or safety velocity fuse. These fuses are an excellent safety device when applied correctly. Refer to U.S. Depart-

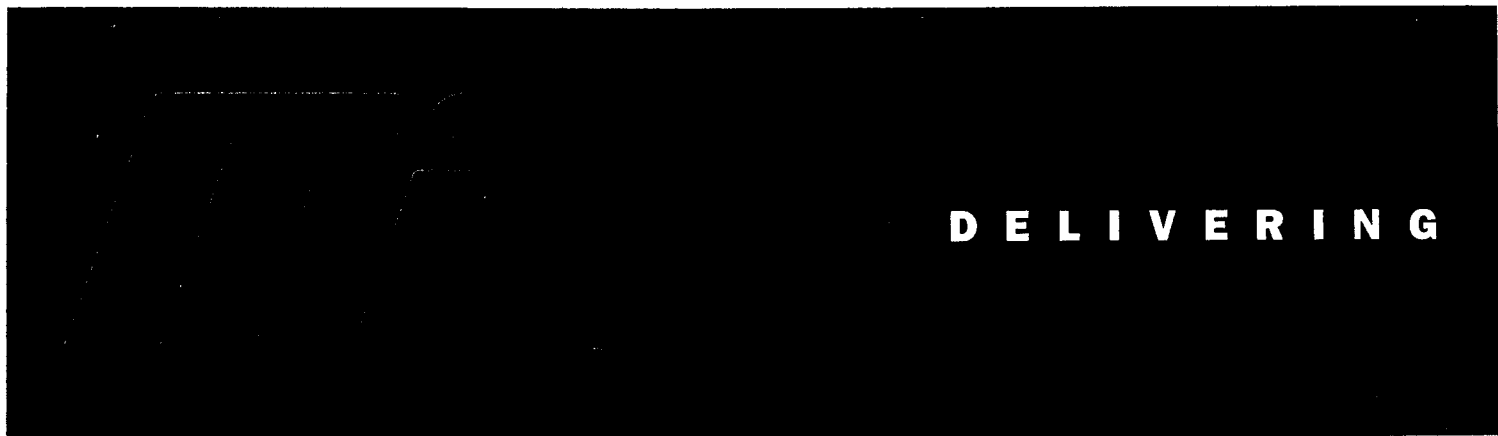
grinding area supposedly required it. Grinding accounted for only 20% of the demand, so 80% of the plant was supplied with air at a much higher pressure than needed. We didn't calculate how much the higher pressure was costing, but intuition says it amounts to thousands of dollars a year.

**Table 2. Off-design performance**

| Design pressure | Actual pressure | Performance loss |
|-----------------|-----------------|------------------|
| 100 psig        | 90 psig         | 7% to 17%        |
| 90 psig         | 80 psig         | 7% to 16%        |
| 80 psig         | 70 psig         | 17%              |
| 70 psig         | 60 psig         | 20%              |
| 100 psig        | 60 psig         | 50%              |
| 90 psig         | 60 psig         | 39%              |
| 80 psig         | 60 psig         | 33%              |

**If the header pressure stays steady, and the process inlet pressure falls, then the restriction is in the feed line from the header to the process.**

Testing with a needle gauge at full operation revealed that the actual inlet pressure to the tool was 63 psig at load, but the header pressure stayed at 98 psig. In other words, there was a 35-psig pressure drop between the header and each grinder. Further testing revealed that the



ment of Labor, Occupational Safety and Health Administration — Power Operated Tools 1926.302, page 2, paragraph 1926.302(b)(7), which mandates a safety velocity fuse on all hoses larger than 1/2 in. inside diameter.

#### A real-world example

More often than not, a process requires some minimum pressure. Trace these so-called requirements to their origin to determine if they are actual OEM specifications or simply an operator's perception.

A recent client was running the plant headers at 100 psig to 110 psig because a critical hand-tool grinding process was believed to require 98 psig to run correctly. Therefore, they reasoned, the system should run at 98 psig or more.

When you hear things like this, dig for more information. If the system header pressure falls below 98 psig, the grinders probably don't work well. Production personnel probably don't know the actual pressure at the tool or how much air the tool uses. The rest of the plant could have run at 80 psig, but it operated at 98 psig because the

grinders only needed 75 psig for optimum performance.

The operators argued that they found the recommended 3/4-in. hose to be too heavy, so they used 3/8-in. hose instead. The smaller hose restricted the air flow, which produced a substantial pressure drop. Furthermore, the 3/8-in. hose used standard quick disconnects, which add-

**Table 3. Pressure drops at flow rates**

| Tool size (hp) | Flow (cfm) | Pressure drop per 50 ft. (psi) |              |
|----------------|------------|--------------------------------|--------------|
|                |            | 1/2 in. hose                   | 3/4 in. hose |
| 1              | 30         | 2.4                            | 0.4          |
| 2              | 60         | 14.9                           | 2.2          |
| 3              | 90         | 41                             | 4.6          |

(Assumes 90 psig supply, does not include fittings)

ed their own 23-psi pressure drop.

We changed the standard 3/8-in. quick disconnects to industrial quick disconnects costing only \$2.50 per pair — a

## HOW QUICK DISCONNECTS WORK

### Lock-ring type with ball-check nipple

- Push the lock-ring coupler to connect. Turn the lock ring about 20° to disconnect. This feature prevents accidental disconnects.
- Nipple with ball check seals the air in the hose or tool connected to the nipple to eliminate blowback and possible uncontrolled hose whip.
- The disconnect will be made under some pressure with variable flow dependent on the installation.
- Flow check-type nipples are more expensive than a standard industrial interchange nipple, which will work in many manufacturer's couplers.

### Exhaust-type

- These quick disconnects use a common standard industrial interchange nipple. When comparing cost, it's important to consider that in many operations, there are usually three or four nipples for every coupling.

- Exhaust-type couplings are push-to-connect, exhaust-style action with a self-locking sleeve to guard against accidental disconnection.
- To connect, push the nipple into the coupler. The locking sleeve slides forward automatically to lock the nipple in place. No air flows through the coupling at this point. Rotate the valve sleeve to open flow and engage the sleeve-lock mechanism.
- To disconnect, rotate the valve sleeve in the other direction to shut off the air flow and vent downstream air to atmosphere. The locking sleeve can then be retracted and the nipple removed.
- The valve sleeve acts as an integral shutoff valve that allows connect and disconnect at zero pressure. The valve sleeve is operated independently of the locking sleeve. When the sleeve is moved to stop air flow, it automatically vents downstream pressure so disconnect can be performed at zero pressure.
- Exhaust couplers eliminate the need for flow-check nipples and still meet safety issues by connecting and disconnecting at zero pressure.

## THE DIFFERENCE.



whopping \$5 per station — to reduce the pressure drop to 5 psig. Then, we replaced the 3/8-in. hose with 1-in. pipe routed to the base of the work stations at a cost of \$30 each. Next, we installed a regulator that delivered full flow to the grinders at 75 psig with 80-psig feed pressure. Finally, we reduced the header pressure to 85 psig. About 18 months later, grinder repair costs had decreased and production throughput increased by 30% with the addition of more equipment. The cost of materials to implement these changes was \$1,362 for nine grinders. Even with the production increase and new equipment, the average total air demand fell from 1,600 to 1,400 cfm.

The key to this success was monitoring the workstation inlet pressure while simultaneously monitoring header pressure. If the header pressure stays steady, and the process inlet pressure falls, then the restriction is in the feed line from the header to the process.

### Break down the connection

This case study demonstrates that small hose represented only 12 psid while the quick disconnects represented 23

psid. Often, but not always, a quick disconnect is the best answer for overall productivity. But, size the quick disconnect for the maximum expected flow and the allowable pressure loss. Read the manufacturer's performance data sheet.

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- Never select by connection size — select by acceptable performance at specified flow and entry pressure.
- If you want to use the same quick disconnect everywhere for flexibility, do it. But, size them for the single

## Measure before you act

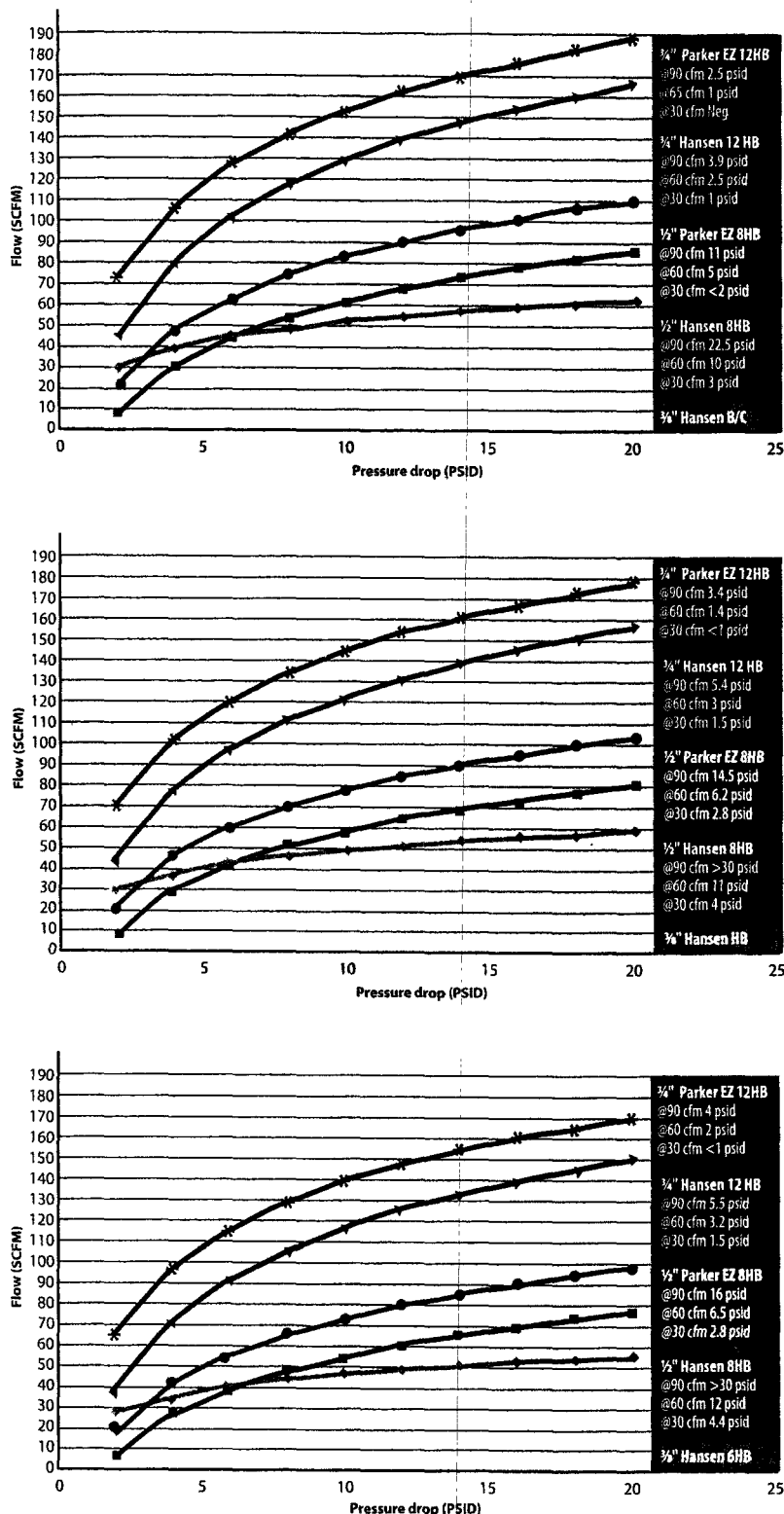


Figure 1. These measured performance characteristics of several quick disconnects show flow as a function of pressure drop.

largest flow demand at the lowest expected pressure.

- Remember that each feed has at least two quick disconnects.

- Use quick disconnects that shut off the flow when disconnected to eliminate potential hose whipping.

- Consider ISO 4414 exhaust-type quick disconnects that bleed off the air trapped inside the connection to eliminate blasting compressed air onto the operator at disconnect. It's easier to uncouple a depressurized fitting.

- Quick disconnects should have proper safety latches, wires and keepers or be of a design that won't open when dragged over the ground, floor or machinery.

## Seek tested performance curves

Don't assume that because couplers appear similar the performance is similar. Review the performance curves or, even better, measure the pressure loss at specific flows. On a recent audit to help select the proper disconnect for a major tool operation, we tested the pressure drop on two specific types of quick disconnect — a lock-ring coupler with a ball-check nipple versus an exhaust-type coupler with a standard nipple. Both had 1 1/4 in. diameter coupler bodies and ports sizes of 3/8 in., 1/2 in. and 3/4 in. The 3/8-in. nipple on the lock-ring type coupler didn't have a ball check to shut off the air. The 1/2-in. and 3/4-in. units did. The exhaust-type couplers had the full shutoff and exhaust to allow disconnect at zero pressure.

Figure 1 shows the three sets of performance curves that reflect the measured pressured drop of each quick disconnect at various flows and inlet pressures. The results will probably vary by manufacturer. The key is to optimize performance by investigating.

We found a significant pressure drop difference between the 1/2-in. quick disconnects. The exhaust coupler could work in an acceptable manner from less than 30 cfm to as much as 60 cfm and still maintain 100 psig inlet with 80 psig to the tool or 90 psig inlet with 70 psig to the tool.



## Did You Know?

35% of all compressor controls or control systems are not operating properly

40% of all air system dryers are installed backwards, plugged, bypassed, or otherwise not performing their intended purpose

50% of all air systems are not piped correctly

80% of all plants could reduce their air demand by one quarter if they eliminated the air use or switched to an alternative

Amazingly, 30% of all brand new or reconstructed air systems are engineered incorrectly

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## PERFORMANCE Compressors

• The 1/2-in. lock ring/ball check nipple quick disconnect appears acceptable at 30 cfm but probably won't be acceptable at 60 cfm.

• The 3/4-in. quick disconnects are closer in performance, but the lock ring/ball check type introduces 30% to 40% more pressure drop.

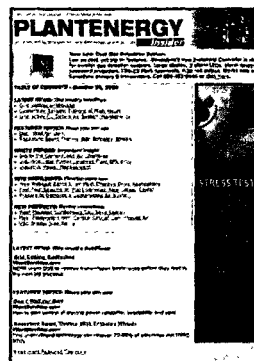
• The 3/8-in. lock ring/ball check nipple quick disconnect tested didn't have the ball check valve in the nipple, which accounts for its lower pressure drop compared to the 1/2-in. lock ring coupler, which did. This, of course, means that the safety feature to control potential hose whip isn't incorporated into the 3/8-in. lock ring set.

This test data isn't intended to recommend one disconnect over another. For the particular application investigated, with many grinders and impact tools using between 60 scfm and 90 scfm, the exhaust-type quick disconnect exhibited the best overall performance and economics. On a different application, testing may well dictate another choice. The important point is to select quick disconnects, hose and pipe with diligence and attention to detail. Although disconnects are a relatively inexpensive piece of equipment, if misapplied, they can be costly. ☺

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